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OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 MAY 1910 NUMBER 5

SPRING MEETING, ATLANTIC CITY, MAY 31-JUNE 3

PROGRAM

Tuesday afternoon and evening, May 31

Informal reunion of members in the parlors of the Marlborough-Blenheim.

Wednesday, June 1, 10 a.m.

PROFESSIONAL SESSION

Business meeting. Reports of Committees, Tellers of Election, New Business.

PAPERS ON MACHINE CONSTRUCTION AND OPERATION

THE SHOCKLESS JARRING MACHINE, Wilfred Lewis.

A COMPARISON OF LATHE HEADSTOCK CHARACTERISTICS, Prof. Walter Rautenstrauch.

THE STRENGTH OF PUNCH AND RIVETER FRAMES MADE OF CAST IRON, Prof. A. L. Jenkins.

Wednesday afternoon and evening

The afternoon is left unassigned to give opportunity for sight-seeing. Roller chairs for the boardwalk will be available for the visiting members and guests through the courtesy of the Local Committee.

In the evening, entertainment on the steel pier has been provided by the committee.

Thursday, June 2, 10 a. m.

PROFESSIONAL SESSION

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THE ELASTIC LIMIT OF MANGANESE AND OTHER BRONZES, J. A. Capp.

THE HYDROSTATIC CHORD, R. D. Johnson.

THE RESISTANCE OF FREIGHT TRAINS, Prof. Edw. C. Schmidt.

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GAS POWER SECTION

BUSINESS MEETING AND REPORTS OF COMMITTEES.

PAPERS

A REGENERATOR CYCLE FOR GAS ENGINES USING SUB-ADIABATIC EXPANSION, Prof. A. J. Frith.

GAS ENGINES FOR DRIVING ALTERNATING CURRENT GENERATORS, H. G. Reist.

TWO PROPOSED UNITS OF POWER, Prof. Wm. T. Magruder.

SOME OPERATING EXPERIENCES WITH A BLAST FURNACE GAS POWER PLANT, H. J. Freyn.

Thursday, 9 p. m.

Reception, followed by conferring of Honorary Membership on Rear-Admiral George W. Melville, U. S. N., Ret. A brief address will be made by Admiral Melville, and the evening will conclude with dancing and refreshments.

Friday, June 3, 10 a. m.

PROFESSIONAL SESSION

PAPERS ON POWER TRANSMISSION

IMPROVEMENTS IN LINESHAFT HANGERS AND BEARINGS, Henry Hess.

EXPERIMENTAL ANALYSIS OF A FRICTION CLUTCH-COUPLING. Prof. Wm. T. Magruder.

AN IMPROVED ABSORPTION DYNAMOMETER, Prof. C. M. Garland.

CRITICAL SPEED CALCULATION, S. H. Weaver.

REPRESENTATIVE OF THE PROFESSION IN PENNSYLVANIA AND NEW JERSEY

For several years the Spring Meeting of the Society has been held in cities where there has been an opportunity of visiting places of interest and inspecting engineering enterprises, and the time of the members has been very fully occupied in taking advantage of such excursions as the generosity and coöperation of the local members have made possible. While these meetings have all been thoroughly enjoyed, it was thought that it would be a welcome change to hold a meeting at a resort where those attending would have more time for renewal of acquaintance and for personal intercourse, instead of devoting so much attention to matters outside of the interests directly related to the Society and its membership. The last meeting of this sort was the Spring Meeting in 1903 held at Saratoga. The present meeting at Atlantic City should be an equally pleasant occasion, since there is no place in the country better adapted for holding a convention and the meeting is held at a time which is one of the most delightful in which to spend a few days on the New Jersey shore.

The Marlborough-Blenheim Hotel, the Convention headquarters, is situated at the central point of Atlantic City's famous seven-mile board walk, and occupies a block and a half on the ocean front looking southward and 200 yards on the City Park looking eastward. It has a capacity of 1100 and makes many provisions for the comfort of its guests, including the open-air plaza and the solariums overlooking the ocean.

LOCAL COMMITTEE

James M. Dodge, *Chairman*

J. Sellers Bancroft
J. C. Brooks
James Christie
Morris L. Cooke
Charles Day
Kern Dodge
Francis H. Easby
Thomas M. Eynon
John Fritz
Harris R. Greene
G. T. Gwilliam
E. P. Haines
Robert E. Hall
Henry Hess
Edward I. H. Howell
Arthur C. Jackson
William C. Kerr
Wilfred Lewis
E. P. Linch
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J. A. C. L. de Trampe
Wm. S. Twining
Mr. Van Gilder
S. M. Vauclain
William R. Webster
Tilden White
Walter Wood

MEETING IN ENGLAND

A program of the joint meeting of The American Society of Mechanical Engineers and the Institution of Mechanical Engineers has been issued by the Institution. As already announced this meeting will be held in Birmingham and London and will begin on Monday, July 25. A local committee consisting of the Right Hon. the LORD MAYOR OF BIRMINGHAM, Alderman W. H. Bowater, together with members of the Institution and other gentlemen resident in the neighborhood, has been formed to make the necessary arrangements. A ladies' committee will be formed in Birmingham to make arrangements for the entertainment of ladies accompanying the members of both societies.

PROVISIONAL BIRMINGHAM PROGRAM

Monday, 25th July. Arrival in Birmingham

Tuesday, 26th July

Morning.—The Right Hon. the Lord Mayor of Birmingham and the Members of the Local Committee will receive and welcome the President, GEORGE WESTINGHOUSE, Esq., and the Officers and Members of the American Society of Mechanical Engineers, and the President, JOHN A. F. ASPINWALL, Esq., and the Council and Members of the Institution of Mechanical Engineers.
READING AND DISCUSSION OF PAPERS.
LUNCHEON in the Town Hall.

Afternoon.—Visits to Stratford-on-Avon, Worcester, Gloucester, or Bournville; and local Works.

Evening.—Garden Fête.

Wednesday, 27th July

Morning.—READING AND DISCUSSION OF PAPERS.
LUNCHEON in the Town Hall.

Afternoon.—Visits to the University and local Works.

Evening.—Reception in the Council House, by invitation of the Right Hon. the Lord Mayor of Birmingham.

Thursday, 28th July

Visits to Works in Coventry and Rugby; also to Warwick, Leamington, Kenilworth, or Lichfield.

PROVISIONAL LONDON PROGRAM

Thursday, 28th July

Evening.—Conversazione at the Institution.

Friday, 29th July

Morning.—Reading and Discussion of Papers.
Afternoon.—Garden Parties at Private Houses.
Evening.—Institution Dinner in the Connaught Rooms, Freemason's Hall, Great Queen Street, W. C. (Including Ladies.)

Saturday, 30th July

Morning and Afternoon.—Excursion by Rail and River to Windsor and Henley.
Evening.—Reception at the Garden Club in the Japan-British Exhibition at the White City.

It is intended that Invitation Cards be handed to the American visitors on their arrival in Birmingham.

The privileges and invitations in connection with the Meeting are *personal* and are *not transferable.*

PRINCIPAL HOTELS CENTRALLY SITUATED IN BIRMINGHAM AND NEIGHBORHOOD

Queen's
Grand
Imperial
Midland
Colonnade
Swan
Plough and Harrow (Hagley Road 1½ miles)
Cobden (Temperance)
Hen and Chickens (Temperance)
Kenilworth (27 miles): King's Head; Manor House
Knowle (10 miles): The Forest
Leamington (23½ miles): Regent; Clarendon: Manor House
Lichfield (18 miles): George; Swan
Stratford (26½ miles): Shakespeare; Red Horse; Red Lion
Warwick (22 miles): Warwick Arms
Wolverhampton (12½ miles): Victoria Star and Garter.

For the convenience of members of The American Society of Mechanical Engineers, who are expected to arrive in Liverpool on Sunday, 24th July, and to proceed by special train to Birmingham on the 25th, a list of the principal hotels in Liverpool and the neighborhood, and in Southport, follows.

PRINCIPAL HOTELS IN LIVERPOOL AND NEIGHBORHOOD

Adelphi, Ranelagh Place
Exchange Station, Tithebarn Street
North Western, Lime Street
Hotel St. George, Lime Street
Angel, Dale Street
Compton, Church Street
Feathers, Clayton Square
Imperial, Lime Street
Stork, Queen Square
Union, Parker Street
Washington, Lime Street
Waterloo, Clayton Square
Laurence's Temperance, Clayton Square
Shaftsbury (Temperance), Mount Pleasant.
Hotel Victoria, New Brighton (2½ miles by ferry boat)
Leasowe Castle Hydro., Wallasey (3¾ miles)
Royal, Waterloo (5 miles)
Blundell Sands, Blundellsands (6 miles)
Royal, Hoylake (7¼ miles)
New Hydro., West Kirby (8½ miles)
Chester (15 miles from Birkenhead: Queen Hotel (opposite Railway Station); Grosvenor Hotel (center of City); Blossoms Hotel
Southport (18½ miles from Liverpool): Prince of Wales. Lord Street; Palace; Royal; Victoria; Waverley; Queen's, all on Promenade.

MEETING IN ST. LOUIS MAY 14

A meeting of the Society will be held in St. Louis, May 14, in which the Engineers Club of St. Louis are to coöperate. The paper will be Freight Train Resistance by Prof. Edward C. Schmidt, which is published in this number of the Journal.

REPORTS

MEETING IN ST. LOUIS APRIL 9

The meeting of the engineers of St. Louis, April 9, was conducted by The American Society of Mechanical Engineers with the coöperation of the St. Louis Section of the American Institute of Electrical Engineers as well as of the Engineers Club of St. Louis A symposium on Electric Drive in the Machine Shop was presented, to which three papers were contributed by the Society: The Economy of the Electric Drive, by A. L. DeLeeuw, Mem. Am. Soc. M. E.; Economical Features of Electric Motor Applications by Charles Robbins, of the Westinghouse Electric and Manufacturing Company, and associate member of the American Institute of Electrical Engineers; Mechanical Features of Electric Driving, by John Riddell, Mem. Am. Soc. M. E. A paper, Selection and Methods of Application of Motors and Controllers, by Charles Fair, of the General Electric Company, a member of the Institute was contributed by the American Institute of Electrical Engineers. The attendance was nearly 100.

MEETING IN NEW YORK APRIL 12

The New York monthly meeting was held Tuesday evening, April 12, in the Auditorium of the Engineering Societies Building, with the American Institute of Electrical Engineers coöperating. The subject was Electric Drive in the Machine Shop with the four papers listed above under the meeting at St. Louis.

This subject of electric driving has long been in preparation with a view to presenting in the papers and discussions the recent developments in electric motor applications to machine tools and the economic features of such applications where motors are installed either for direct driving or in connection with lineshafts for group driving. The economic side was very fully discussed by representatives of the machine tool industry ando ther users of motors as well as by the motor manufacturers. Mr. Fred. L. Eberhardt, Vice-President of the National Machine Tool Builders Association, spoke officially for his

organization of their efforts, with the American Association of Electric Motor Manufacturers, for the securing of standards for motor equipment. Mr. A. L. DeLeeuw, a member of the committee of the National Machine Tool Builders Association to consider this subject, followed with a detailed account of the efforts at standardization to date, in which he said fifteen points had been raised for discussion and that seven of them had been considered by the joint committee thus far, among them being the subjects of horsepower, voltages, speeds and ratings.

The papers were also discussed by Henry Hess, of the Hess-Bright Manufacturing Company, Philadelphia, Pa.; L. R. Pomeroy, of the Safety Car Heating & Lighting Company, New York; Gano Dunn, Vice-President of the Crocker-Wheeler Company, Ampere, N. J.; Charles Day, of Dodge & Day, Philadelphia; W. S. Rogers of the Bantam Anti-Friction Company, Bantam, Conn.; Carl G. Barth, Philadelphia; and H. A. Horner, Philadelphia.

MEETING OF THE COUNCIL

A meeting of the Council was called to order in the rooms of the Society, April 12, 1910. Present, Messrs. George M. Bond, Chas, Whiting Baker, J. Sellers Bancroft, H. L. Gantt, James Hartness, Charles Wallace Hunt, F. R. Hutton, I. E. Moultrop, E. D. Meier, H. G. Reist, Frederick W. Taylor, Jesse M. Smith and the Secretary. In the absence of the President, Col. E. D. Meier took the chair.

The Secretary announced the deaths of James Blessing and Gardiner C. Sims.

The resignations of A. E. Coleman, Jr., Zareh H. Kevorkian, George E. Kirk, J. E. Tatnall and Ephraim Smith were read and accepted, and the membership of Thomas M. Keith, George L. Holmes, Barton H. Cameron, Rafael de la Mora, W. Allen Pendry, Edward S. Seaver and Charles L. Weil was declared to have lapsed.

Voted: To accept the report of the special committee appointed by the Council to go to St. Louis, and to express the very hearty appreciation of the Council; and to refer the report to the Executive Committee.

The Executive Committee reported as the total booking on the Celtic, to date, of those who will attend the Joint Meeting in England: 144 members and ladies; going by other routes or already in Europe 81 making a grand total of 225.

Voted: To appoint as a Committee on Arrangements, in con-

nection with the joint meeting in England, Ambrose Swasey, *Chairman*, Charles Whiting Baker, *Vice-Chairman*, Dr. W. F. M. Goss, George M. Brill, John R. Freeman, and, ex-officio George Westinghouse, President, William H. Wiley, Treasurer, F. R. Hutton, Honorary Secretary, Willis E. Hall, Chairman Meetings Committee, and Calvin W. Rice, Secretary.

The Secretary reported that fifteen members of the Council and Past-Presidents expected to attend the dinner to by given by President Aspinwall, on the evening of Monday, July, 25.

Voted: That Charles Whiting Baker be appointed Honorary Vice-President, to represent the Society at the International Congress of Mining Metallurgy, Applied Mechanics and Practical Geology. The Secretary also presented to Mr. Baker the appointment from the State Department as delegate from the United States.

Voted: To appoint Worcester R. Warner Chairman of the Committee on Land Fund.

Voted: To refer to the Executive Committee, with power, in the matter of coöperation with the Verein Deutscher Ingenieure in the preparation of biographies of eminent engineers.

Voted: To approve the applications for a Student Branch at the University of Arkansas, Fayetteville, Ark.

Voted: To adopt the following amendments:

ELECTION OF MEMBERS

B 11 Each person elected to membership, except an Honorary Member must subscribe to the Constitution, By-Laws, and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society. Resignations from membership shall be presented to the Council for action.

FEES AND DUES

B 16 The initiation fee and the annual dues for the first year shall be due and payble on the first day of the month following the date of the election of a Member, Associate, or Junior. The annual dues for each ensuing year shall be due and payable in advance on the corresponding day in each year thereafter.

Upon the payment of the initiation fee and the annual dues for the first year the person elected shall be entitled to the rights and privileges of membership in the grade to which he was elected. The date of payment of a member's annual dues may be changed to the first day of any other month, and a *pro-rata* adjustment of the dues made, by application to the Secretary.

B17 A Member, Associate or Junior in arrears for dues for one year, on the first day of October previous to the annual Meeting, shall not be entitled to vote, or to receive the transactions or the publications issued by the Society

thereafter until such dues have been paid. Should the arrears for dues or otherwise be for more than two years, the name of such person shall be presented to the Council for such action as it deems advisable under C 24. Should the right to vote, or to receive the publications of the Society be questioned, the books of the Society shall be conclusive evidence.

B 18 The council may, in its discretion, restore to membership any person dropped from the roll for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

Voted: That the Council accept the invitation of the National Steam and Hot Water Fitters Association to a conference leading to the adoption of uniform standards for flanged and screwed cast-iron fittings, and that the Secretary communicate with the various members of the Council and specialists in power-house practice and ask their suggestions for names for such a committee.

Voted: That a message of congratulation and greeting be sent to the Aero Club of American on the opening of the club rooms on the evening of Wednesday, April 13, in the Engineering Societies Building.

Voted: That Charles Whiting Baker be appointed Chairman of a Committee, with power to increase the number to investigate the matter of a proposed bill now before the legislature to license engineers, and to report to the Council at its next meeting.

REQUEST FOR 1903 YEAR BOOK

A copy of the Year Book for 1903 is needed to complete the files of the Society. Any member willing to furnish a copy will please communicate with Calvin W. Rice, Secretary, at the rooms of the Society.

STUDENT BRANCHES

PENNSYLVANIA STATE COLLEGE

At the March meeting of the section held March 16, the topic for the evening was Methods of Coal Mining, which was ably handled by George B. Wharen, A. F. Goyne and Roy B. Fehr (1910). The papers were supplemented by views of mines and mine apparatus thrown on the screen. At the April meeting, Refrigeration and Cold Storage will be discussed.

PURDUE UNIVERSITY

On March 24, F. H. Clark, Genl. Supt. M.P. of the C. B. & Q. R. R. addressed the student section of Purdue University on The Functions and Work of the Motive Power Department, followed by a general discussion in which many points of interest were enlarged upon. Professor Ensley, of the university, addressed the meeting on April 6, on Recent Developments in Brake Shoe Tests, on whichhe is an authority. His address was supplemented by lantern slides.

UNIVERSITY OF CINCINNATI

The student branch of the University of Cincinnati had as the speaker at its meeting on March 25, James B. Stanwood, Mem.Am. Soc.M.E., who presented a very interesting paper on The Development of Non-Condensing Engines. Harry M. Lane, Mem.Am.Soc. M.E., gave a discussion of the paper. At the meeting on April 15, William Goodman, manager of Laidlaw-Dunn-Gordon Company, presented a paper on Air Compressors and their Manufacture.

WISCONSIN UNIVERSITY

At the April meeting of the section, G. A. Glick (1910) presented a paper on A 15,000-kw. Steam Engine Turbine Plant, based on Mr. Stott's paper published in The Journal, March issue. The paper was followed by a discussion in which Assistant Professor A. G. Christie told of some of the difficulties encountered in testing the engines referred to in the paper. The following officers were elected; President, John S. Langwill; Vice-President, Henry A. Christie; Corresponding Secretary, Karl L. Kraatz; Assistant Secretary, Guy H. Suhs; Treasurer, Angus MacArthur.

OTHER SOCIETIES

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

Charles Whiting Baker, Vice-President of the Society, has been appointed Honorary Vice-President to represent the Society at the International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, to be held at Düsseldorf, Germany June 20–23, 1910. Mr. Baker has also received appointment from the State Department as delegate from the United States.

BOSTON SOCIETY OF ARCHITECTS

At the dinner of the Boston Society of Architects, held at the Parker House, April 1, 1910, Calvin W. Rice, Secretary Am. Soc.M.E., was the guest of honor. The topic for consideration was Office Organization. Mr. Shreave of Carrere & Hastings was also a guest and spoke from knowledge not only of the office with which he is connected, but also of that of McKim, Mead & White.

Mr. Rice took occasion to explain the necessity of organization in the office of the society, as in a business, for the reason that members of the society who are business men expect efficient service whenever they communicate with the society on any matter. The Society is essentially an organization of trained men. The variety of the inquiries, several hundred a day, also requires a complete staff, and it must be organized if useful and practical attention is to be given these inquiries. The most important idea in connection with an engineering society is that, in a larger sense than the individual, it must serve the profession; rather than that it is simply an aggregation of persons for selfish interests. In other words, the association must be organized for progressive and helpful work. Such is the organization of The American Society of Mechanical Engineers.

AMERICAN ELECTROCHEMICAL SOCIETY

In response to a request, the Secretary has sent to Dr. Jos. W. Richards, South Bethlehem, Pa., secretary of the American Electrochemical Society, a list of members of this Society resident in the Pittsburg district, who are to be specially invited to the convention to be held at Pittsburg, May 5–7. C. E. Foster, Mem.Am.Soc.M.E., and John Brashear, Hon.Mem.Am.Soc.M.E., are among the speakers. Further announcement will be found on a later page of The Journal.

NECROLOGY

WILLIAM WILBERFORCE CHURCHILL

William Wilberforce Churchill died at Oshkosh, Wis., on March 24, 1910. He was born at Monroe, Wis., January 6, 1867, and was the son of Norman and Dr. Ann Sherman Churchill. After graduation from the Monroe High School in 1883, he spent one year at Rose Polytechnic Institute, and in 1886 entered Cornell University, from which he was graduated in 1889 with the degree of M.E. He was made a Fellow of Sibley College for 1889–1890, and received the degree of M.M.E. in 1890.

After graduation Mr. Churchill spent a few months with E. P. Allis & Co., Milwaukee, Wis. In 1890 he entered the employ of Westinghouse, Church, Kerr & Co., where he remained until his retirement because of a breakdown in health, in 1906. He rose through various intermediate positions in Chicago, Pittsburg and Boston to be chief mechanical engineer of the company's headquarters in New York. At the time of his retirement he was vice-president and director in the company. During his sixteen years of service he superintended the construction of the Boston terminal; the Kingsbridge power house, New York City; the Atlanta water plant, Georgia; the Lackawanna & Wyoming Valley R. R., Pennsylvania; The Grand Rapids, Grand Haven & Muskegon R.R.; Hotel Pontchartrain, Detroit, Mich.; the Northern Colorado Power Company, Denver, Colo.; the electrification of the Long Island R. R., New York, and many others. In 1902 he spent some time in Europe in connection with the electrification of the London Underground Railway.

Mr. Churchill was a member of the New York Railroad Club, the Cornell University Club, New York, the American Association for the Advancement of Science, and several Masonic orders.

GARDINER C. SIMS

Gardiner C. Sims, president of the William A. Harris Steam Engine Company, died at his home in Providence, R. I., on March 20, 1910. Mr. Sims was born in Niagara Falls, N. Y., July 31, 1845, and was educated there in the public schools. He began his engineering career with a four years' apprenticeship at the locomotive works of the N. Y. C. & H. R. R. R., West Albany, N. Y., afterward entering the navy yard at Brooklyn, N. Y., but returning to his former employers after three years, to become their chief draftsman. He next became

superintendent of the J. C. Hoadley Engine Works at Lawrence, Mass. Here he met Pardon Armington, with whom he formed a partnership for the manufacture of steam engines, both men devoting their ntire time to experimental work as a result of which they gave to the world the quick-running engine, in opposition to the established engineering practice and precedents. They built the first successful engine for Thomas A. Edison, which was sent to the Paris Exposition with his first dynamo, in 1881.

In 1876 Mr. Sims spent eight months at the Centennial Exposition and was appointed Democratic Commissioner from the State of Rhode Island to the World's Columbian Exposition in 1892, where he was made chairman of the Exposition committee on electricity, electric and pneumatic appliances, and was a member of the committee on machinery and transportation.

At the outbreak of the war with Spain, Mr. Sims volunteered, and was appointed Chief Engineer by the Navy Department and ordered to the navy yard at Boston. For his work in this branch of the service Mr. Sims was made a Lieutenant Commander and received congratulatory letters from Secretary Long and Engineer-in-Chief George W. Melville.

At the close of the war he was summoned by the War Department to assume the position of Superintendent Engineer of the United States Army Transport Service, and discharged his duties with honor until the completion of the work. He was appointed police commissioner in 1902, and at the time of his death was connected with the William A. Harris Steam Engine Company.

HARRY S. HASKINS

Harry S. Haskins, Associate Member of the Society, died at his home in Philadelphia on March 13, 1910. Mr. Haskins was born in Moretown, Vt., March 5, 1834, and at the age of twelve entered the machinists trade, first with Edwin Harrington and later with the Junction Shop, both in Worcester, Mass., where his family had moved. When Mr. Harrington went to Philadelphia, to engage in the building of machine tools, Mr. Haskins accompanied him, and soon afterwards the partnership of Harrington & Haskins was formed, which later became the firm of Edwin Harrington, Sons & Co. On the death of Mr. Harrington, the business became incorporated, with Mr. Haskins as president, an office which he retained until the time of his retirement, in 1900. Throug his mechanical ability and inventive faculty he added many improvements to the gear-cutting machines, hoists and overhanging railways manufactured by the firm.

OPERATING EXPERIENCES WITH A BLAST FURNACE GAS POWER PLANT

By Heinrich J. Freyn

Member of the Society

The use of blast furnace gas engines in this country was first undertaken in 1902 by the Lackawanna Steel Company at Buffalo, followed four years later by the United States Steel Corporation in several of its plants. The import of the problem of utilizing the surplus gas may be realized by considering that eleven million tons of pig iron were produced in 1909 by the United States Steel Corporation, and that for each ton of iron made per day, 25 b.h.p. is available for purposes outside of the power requirements of the blast furnaces, provided this power is itself produced in gas engines. If, therefore, all the blast furnaces of the corporation were blown by gas blowing-engines and all other furnace requirements furnished by gas-electric engines, 750,000 b.h.p. would be available for other purposes.

2 In 1907 there were installed at one of the largest steel plants in this country, four Allis-Chalmers double-acting, four-cycle, twin tandem gas engines, gas cylinders, 42-in. diameter, 54-in. stroke, operating 2000-kw., 25 cycle, 3-phase, 2200-volt, alternating-current generators at 83.3 r.p.m. This addition to the existing steam-electric equipment was completed in 1908 and the electric power produced by these gas engines is used for electric-driven rolling mills and general light and power purposes. The gas-driven generators operate parallel with the adjacent steam units and with other gas-driven generators located 20 miles away.

3 The gas engine plant under discussion has been in regular service for one and one-half years, during which period the experiences and results described in the following pages were obtained.[1] These

[1]These experiences and records were compiled with the able assistance of Mr. Chas. C. Sampson, Mem.Am.Soc.M.E.

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

were not gathered from the indications of one single experiment, or of a series of carefully prepared and conducted tests, but represent the average results of daily observations extending over one year's time. Since the degree of correctness depends on the accuracy of observation and care in recording the results of persons having various degrees of practical and technical training, inaccuracies entailing puzzling inconsistencies may have crept in and the data presented may not in every instance withstand the test of searching criticism; nevertheless it is believed that such information, derived from actual operation, will prove of more interest to the engineering profession than unassailable data obtained under test conditions.

4 The gas supply for the operation of all the gas engines of the plant is taken from six blast furnaces, all of which in 1909 were blown by steam blowing-engines, while the electric power for the plant was derived from both steam and gas-driven generators. This plant is therefore a so-called "mixed" plant, so far as the generation of power is concerned.

5 The quantity of blast furnace gas available for the operation of gas engines was therefore considerably less than it will be when four of these furnaces are blown by gas blowing-engines. Because of the general business depression at the beginning of 1909, only three, and in the months of March and April only two, furnaces were in blast. Normal conditions were resumed about May or June, while all six furnaces were in blast during the months of September and October only.

CONDITIONS OF INSTALLATION

6 The gas power station in question was conceived in 1905, and all preliminary calculations relative to the amount of gas available for operating gas engines were naturally based on the conditions existing at that time, making the proper deductions for reduced gas production due to the furnaces being out of blast for relining. Careful investigations showed that in 1906, 10 per cent of the total gas produced by six blast furnaces, equivalent to 10,200 kw., was available for use in gas-electric engines. The installation of 8000 kw. in gas engines seemed therefore fully warranted, particularly as it was expected that two gas blowing-engines simultaneously ordered would be in operation after November 1907, in which case the gas surplus, even with only five furnaces in blast (one furnace down for relining) would have been more than ample to operate four 2000 kw. of gas-electric units. It could not be foreseen that business conditions would

change so radically in 1908, nor that the two gas blowing-engines would be so delayed, that for three years the saving of gas, which would have materially improved conditions for the electric units, could not be realized.

7 While the logical way to begin would have been with the installation of a number of gas blowing-engines, instead of first taking gas-electric engines in operation, increasing instead of reducing the available quantity of blast-furnace gas, such a procedure was impossible because of the immediate demand for increased electric power created by the installation of new electrically operated mills, as well as on account of local conditions of steam supply for the furnaces, which at that time prohibited the removal of a large boiler house, now occupied by the new gas blowing-engines. From the circumstances, however, that gas-electric engines were installed before any gas blowing-engine equipment existed, and that this power plant, as it so happened, had to be operated for almost two years under the most unfavorable and exacting conditions, a great deal of most valuable experience was gained, in that it was found that such a power plant could be maintained in operation—although with interruptions—with only two furnaces, and for a short period even with only one furnace in blast.

OUTPUT OF POWER PLANT

8 In Appendix No. 1, Table 1 shows the average kilowatt-hour produced by the gas power plant for each month of 1909, from which it appears that the average for the year was 5760 kw-hr., or 72 per cent of the total capacity of the plant, and this average for various months varied from 66.5% to 74% with two furnaces, 61.5% to 80.5% with three, 69% for four, 64% to 82.5% with five, 68.5% to 78% with six furnaces in blast. While during the first few months the number of furnaces in blast was very limited, the total output of the station was nevertheless not affected very materially. In fact, in the month of March, when only furnaces No. 1 and No. 2 were in blast, 74% of the total capacity of the plant was produced, a higher figure than the average output of the plant for the whole year.

SHUTDOWNS AND TIME LOST IN OPERATION

9 A record is being kept as correctly as practicable of all shutdowns and their causes. Table 2, in Appendix No. 1, gives the monthly averages, as well as the percentages of operating time and of time lost chargeable to the engines, and due to outside causes. The

power station is considered in this table as one unit, and the figures are averages applying to the four engines.

10 In Appendix No. 1, Table 2, is given the average monthly operating time of the station for the whole year, from which it appears that the average for the year was 77% of the total possible time, 14.2 % and 8.8% being the percentages of the time lost due to engine repairs and to outside causes beyond control. The respective figures for the first four months of the year show that the operating time from January to April was much lower than during the rest of the year, the lowest figure being 57% in January. However, the month of March with 71% shows again the noteworthy fact that with only two furnaces in blast the operating time was higher than the average for the first half of the year. This was made possible only by shutting off the boiler houses almost entirely. With the boiler houses off, blast furnace No. 1 made more gas than the gas engines, the stoves and one small boiler house could use, so that one bleeder at the furnace had to be kept open. During casting periods the engines were operated on the gas tank. In this manner operation was kept up for one whole week. The time lost chargeable to the engines is considerable for the first four months, due to the fact that owing to the uncertainty of sufficient gas supply under the existing conditions of furnace operation certain repairs and alterations were made on the engines, which otherwise would have been distributed over a longer period of time. It is to be noted that any time lost is rigorously charged against the engines if the latter for any reason are not ready to resume operation at any moment.

11 The time lost due to outside causes was particularly heavy in the first four months of the year, varying from $11\frac{1}{2}$% in April to $19\frac{1}{2}$% in February. In the records the lost time chargeable to outside causes is subdivided into losses due to operation of the plant, such as line troubles, or output not required, etc.; and losses due to lack of gas. This particular information is given in Table 3 of Appendix 1, wherein the plant is again considered as one unit. Shortage of gas was responsible to the greatest extent for lost time from outside causes in the first four months of the year. In January this figure was as high as 94.5 %, and while the average for the first half of 1909 exceeds 60%, the corresponding figure for the second half of 1909 is only 3%.

CONSIDERATIONS OF SAFETY WHEN THERE IS SHORTAGE OF GAS

12 Although the difficulties which were experienced in this period by the frequent inability of the blast furnaces to supply sufficient gas

to maintain operation of the whole plant, without heavily firing coal under the boilers, had no serious effect on the gas power plant, one important question was in the minds of all during this period, namely the safety of the installation. Antedating only a few months this period of gas shortage, a serious accident had happened at another plant where through lack of gas while only one furnace was in blast, the preliminary cleaning plant, the gas holder and parts of the pipe line conveying gas to the engines, exploded with disastrous effect. This accident caused a great deal of uneasiness and alarm in other gas engine plants where several furnaces were out of blast.

13 A gas power plant is endangered in two ways by lack of gas, either from collapsing of the gas holder bell or from explosion. In modern gas-cleaning installations, the so-called secondary washing plant, which refines the gas for use in engines, is usually equipped with some kind of rotary washers. Certain washers of this type, such as the Theisen, can give a vacuum of 3 in. of water, and a discharge pressure from 8 in. to 10 in. higher than the positive or negative pressure on the suction side. The washers deliver the gas to a gas holder under variable pressure dependent upon the raw gas pressure, while it is the principal object of the gas holder to maintain a constant gas pressure at the gas engines, irrespective of what the pressure at the blast furnaces or in the gas-cleaning plant happens to be. As long as the pressure of the gas, and therefore its quantity, is sufficient to allow the rotary washers to keep the gas-holder bell floating; in other words as long as balance exists between the demand for gas on the part of the engines and the supply from the furnaces, there is no danger to the installation.

14 If the gas supply falls below the demand, the volume of gas in the holder will cover the shortage within the limit of its capacity and until the bell, descending completely, rests on its landing beams. The rotary gas washers will then continue to operate, creating a depression in the gas conduits by which they are connected to the gas main at the furnaces. The latter is virtually a large gas receiver into which all blast furnaces discharge their gas, and which in turn supplies the hot-blast stoves, the boilers, and the gas-cleaning plant. The vacuum created by the rotary washers will naturally be communicated to this main gas flue, but cannot be maintained as the overhead flue is connected with the atmosphere through hot-blast stoves and boiler stacks. Air will therefore rush into this flue and into the gas-cleaning plant and be drawn into the rotary washers together with whatever gas is supplied, and discharged into the gas holder.

As long as these conditions exist, the gas-holder bell is not in danger from collapsing, but there is imminent danger from explosion to the whole plant. Rotary gas washers do not discriminate between gas and air, and continue to operate, filling all gas flues, gas holder and engine connections with a mixture of gas and air which under certain conditions is highly explosive. Should backfiring occur in the gas engines when receiving a mixture of gas and air instead of pure gas; that is, should the fresh incoming charge accidentally be ignited, consequences would be as prompt as disastrous—the air and gas mixture in the pipe system would explode, possibly wrecking the whole installation by a series of explosions.

15 This is precisely what did happen in the accident mentioned, and profiting by this experience steps were taken to prevent the occurrence of such an accident at the plant under discussion. Power house, gas washing plant and blast furnace office were connected by two independent telephone lines, and recording instruments, in addition to ordinary U-tubes, were installed in the washer building and at the blast furnace office, so that not only may the gas pressure be observed at any time, but it is automatically recorded for each period of 24 hours. Moreover an automatic alarm was installed at the blast furnace office, which rings a bell as soon as the gas pressure in the raw gas descends below a certain danger point, and whistle signals operated by solenoids from the blast furnace office were provided in the boiler house to inform the head fireman of the number of boilers to be "taken off" or put on gas. In addition an automatic bell was placed in this boiler house, calling the operators' attention to any drop below normal in the gas pressure.

16 Independently of the blast furnace department, the gas-cleaning plant operators were also carefully watching the gas pressure. The position of the gas-holder bell was made visible at any time by a system of incandescent lamps in the washerhouse, and strict orders regarding the use of the gas were issued by the blast furnace superintendent, instructing the men to favor the gas engines under any circumstances, as it was fully recognized that having taken care of the requirements of the hot-blast stoves, the remaining gas could not possibly be more efficiently utilized than in the gas engines. The practice was to shut off the gas immediately at a certain number of gas-fired boilers, as soon as the pressure in the overhead gas flue dropped below a predetermined point. Additional boilers were taken off if the gas pressure did not recover, so that sometimes as many as 24 boilers were being fired by coal exclusively. If this did not have

the desired result, stoves were taken off for short periods to increase the gas pressure above the danger point. At last, if all these steps did not improve the situation, one or more gas engines were shut down.

17 Fortunately in the majority of cases the blast furnace operators know in advance if the gas supply is likely to fail, and communication could easily be established to warn the departments concerned of the impending gas shortage. The diagram, Fig. 1, plotted from a

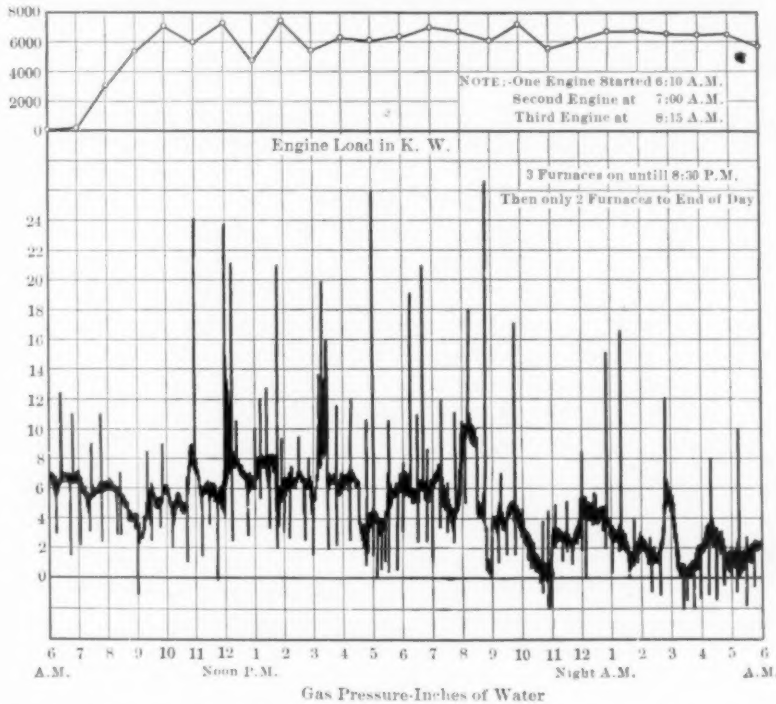


FIG. 1 DIAGRAM PLOTTED FROM BRISTOL CHART SHOWING GAS PRESSURE WHERE THE GAS ENTERS THE GAS-CLEANING PLANT

Bristol chart representing a 24-hour record of the gas pressure at the point where the gas enters the gas-cleaning plant, is a good illustration of the conditions existing on many occasions.

18 The system of close observation and of coöperation among the departments concerned worked to perfection, but nevertheless conditions existed at times which with all due optimism had to be called dangerous. It was frequently necessary to keep several gas engines running, with the gas pressure dropping below the danger point

momentarily or even for periods of a few minutes. This was unavoidable if the operation of certain departments dependent upon a supply of electric power was to be maintained with any regularity. If the gas engines had been shut down every time a momentary drop in pressure occurred, it would often have meant an endless amount of shutting down and starting of engines, altogether too frequent for satisfactory operation of the mills, and physically impossible for the gas engine operators.

19 The question whether an automatic safety device should be installed at the power plant under discussion was thoroughly considered and such devices were investigated; but it was decided that the installation of costly safety appliances, which were certain to become inoperative with the normal number of furnaces in blast, was not warranted, as the conditions of gas shortage were exceptional and of temporary nature only. Besides, automatic safety devices, no matter how ingeniously designed, are never "foolproof," and have the reputation of operating without cause and of failing to act when needed. A further drawback is the tendency to over-confidence in the infallibility of a safety device. In this respect gas-cleaning plants should be classed with boiler plants, where the "human element" cannot be eliminated, and safety depends ultimately upon the rigid enforcement of certain established regulations. Responsible operators can use good judgment which automatic safety devices do not possess to decide whether shutdowns are necessary when low gas pressure occurs, possibly for a moment only. This was proved time and time again at the plant under discussion.

20 If such safety devices are considered necessary, however, the arrangement of automatic circuit breakers to shut off the power at the rotary washers, and simultaneously interrupt the ignition circuit of the gas engines, is decidedly better for safeguarding the plant than the installation of butterfly or check valves between rotary washers and gas holder, which shut off delivery under the control of the gas pressure. While in both cases the aspiration of air by the rotary washers is effectively prevented, the former device protects not only the gas cleaning plant but also the gas holder, while the latter may cause collapsing of the holder bell by isolating it from the gas supply.

QUANTITY AND QUALITY OF GAS SUPPLIED TO ENGINES

21 The amount of gas produced by each blast furnace is calculated and distributed in proper proportion among the different places of its consumption. Monthly gas-distribution sheets give a record of

the average daily tonnage of each furnace, the kind of blast, whether natural or dry, the kind of coke used and the coke consumption per ton of iron; further, the average gas analysis for each furnace based on daily determinations of continuous 24-hour samples, the heat value per cubic foot at 62 deg. fahr. and including the sensible heat of the gas at 500 deg. fahr., the temperature of the air at the blowing engines, the number of cubic feet of air blown per minute, and the average blast pressure. From these data the quantity of gas produced by each furnace per minute is calculated according to methods given in Appendix No. 2. The distribution of the gas from one blast furnace based on such calculations is given in the accompanying table (Table 1 herewith), reproduced from data given in Appendix No. 2.

TABLE 1 DISTRIBUTION OF GAS FROM BLAST FURNACE NO. 6
AUGUST, 1909

	Million B.t.u.	Per Cent
Total Gas Generated	324.1	100
Stoves and leakage	130.0	40.
Blowing engines	92.1	28.4
Used at furnace	9.0	2.8
Auxiliaries	4.6	1.4
Total used for blast furnace operation	235.7	72.6
B.t.u. surplus for furnace	88.4	27.4
B.h.p. equivalent of surplus	1470	

22 An excellent practical indicator of the gas quantity available for engine operation is the gas pressure at the cleaning plant. With more than three furnaces in blast the pressure is always sufficiently high to make operation of the gas power station perfectly safe. Fig. 2 shows the average monthly gas pressure at the main water seal

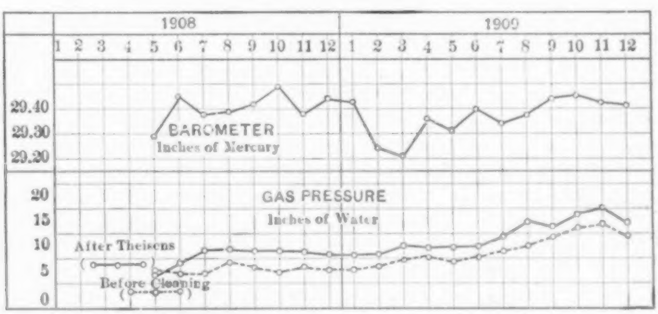


FIG. 2 GAS PRESSURE CURVES—BAROMETRIC PRESSURE (MONTHLY AVERAGES)

where the gas enters the cleaning plant and in the fine gas main after the secondary washers.

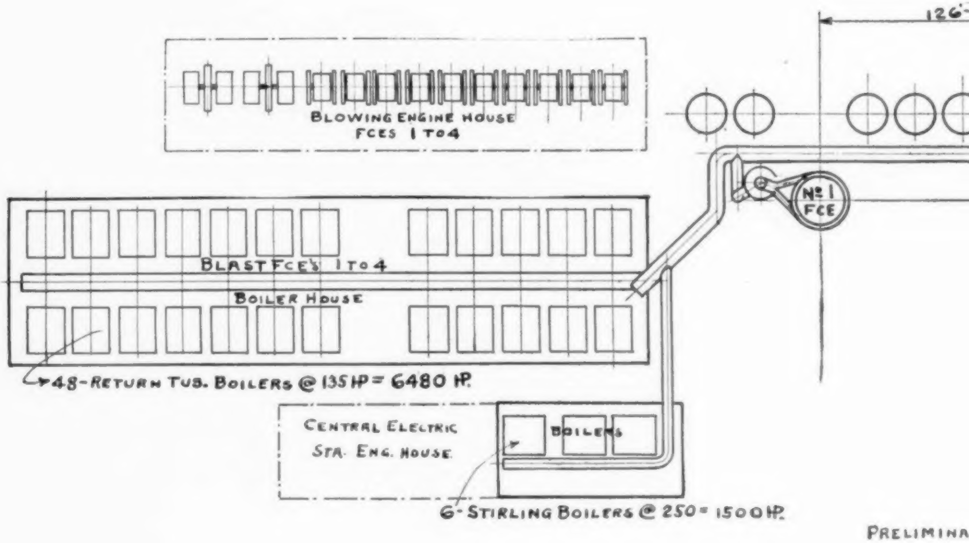
23 While the quantity of blast furnace gas was subjected to considerable variation due to the generally unfavorable conditions which existed in the early part of 1909, the quality of the gas was also found to vary materially, so far as its chemical composition and heat value were concerned. These were influenced not only by changes in the furnace burden and by the kind of product, whether basic iron, Bessemer iron, spiegeleisen or ferrosilicon, but by other causes to which variations frequently recorded from hour to hour, to a large extent could be traced. The blast furnaces discharge their gas into one common overhead gas main supplying stoves, boiler houses and gas engines. The intake for the gas-cleaning plant is located between furnaces No. 2 and No. 3, dividing the total length of the overhead flue in the proportion of one to two, approximately. In view of this central location of the intake nozzle it was expected that by mixing the gas from these furnaces a fairly uniform quality, representing the average of all six furnaces, would be obtained for engine operation. Due to the location of the boiler houses, however, this uniformity of mixture could not be realized.

24 Fig. 3 shows the existing conditions previous to May 1909, and before the boiler plant for four blast furnaces was abandoned in order to make room for the new gas blowing-engine house. The tall boiler stacks caused a flow of gas from furnaces No. 1 and No. 2 to the boiler house situated at the west end of the flue, while the gas from furnaces Nos. 4, 5 and 6 went to two large boiler houses located at the opposite extremity. The gas-cleaning plant received, therefore, almost exclusively gas from furnaces No. 2 and No. 3 or from No. 3 alone, while No. 2 was out of blast. This was proved beyond any doubt by frequently comparing the chemical analysis of the gas delivered at the power station with the analyses of the gas of the individual furnaces.

25 Thus for the month of June 1908 the average composition of the gas from blast furnaces Nos. 1, 3 and 4, which are in close proximity to the gas-cleaning plant intake, was as follows (blast furnace No. 2 being out of blast and blast furnace No. 4 on spiegel):

Blast furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.
1	13.5	25.0	3.4	1.85	93.3
3	12.4	26.7	3.2	2.15	98.1
4	5.1	32.1	3.3	6.30	115.2

FOLDER No. 1


BLOWING ENGINE HOUSE
FCES 1 TO 4
No. 1 FCE
BLAST FCE'S 1 TO 4
BOILER HOUSE
48-RETURN TUB. BOILERS @ 135 HP = 6480 HP.
CENTRAL ELECTRIC
STA. ENG. HOUSE
BOILERS
6-STIRLING BOILERS @ 250 = 1500 HP.

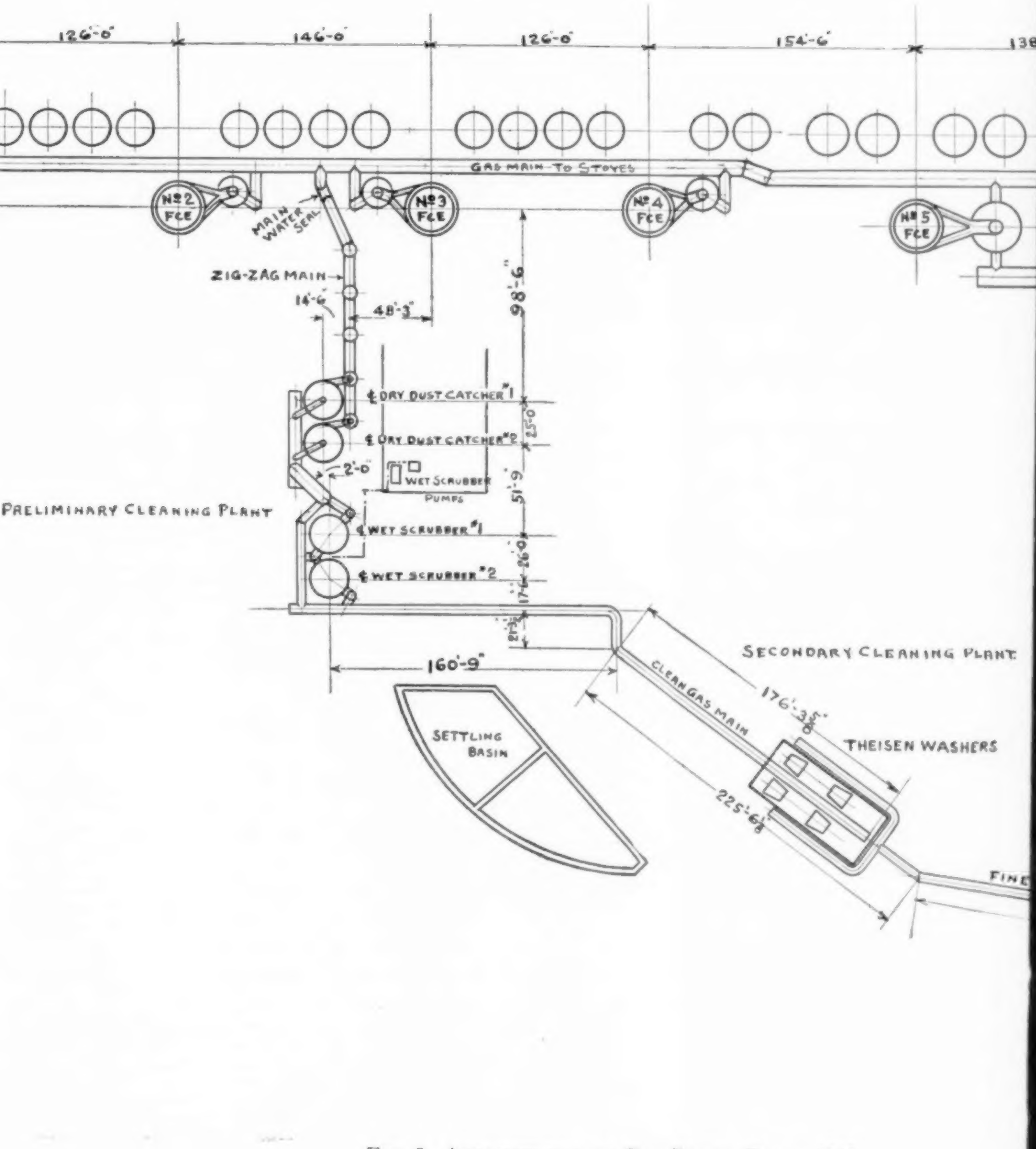


FIG. 3 ARRANGEMENT OF GAS POWER PLANT 1908

NO. 6

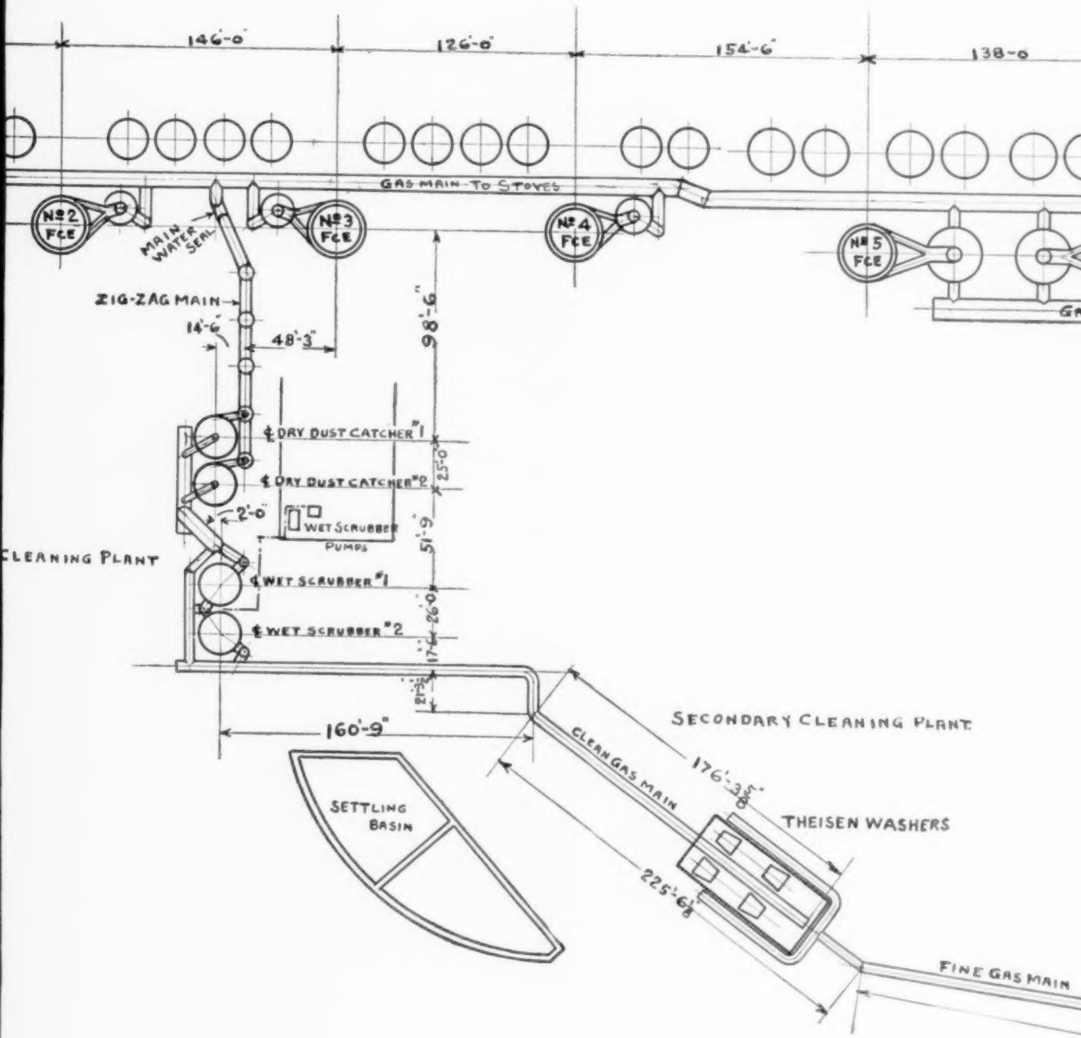
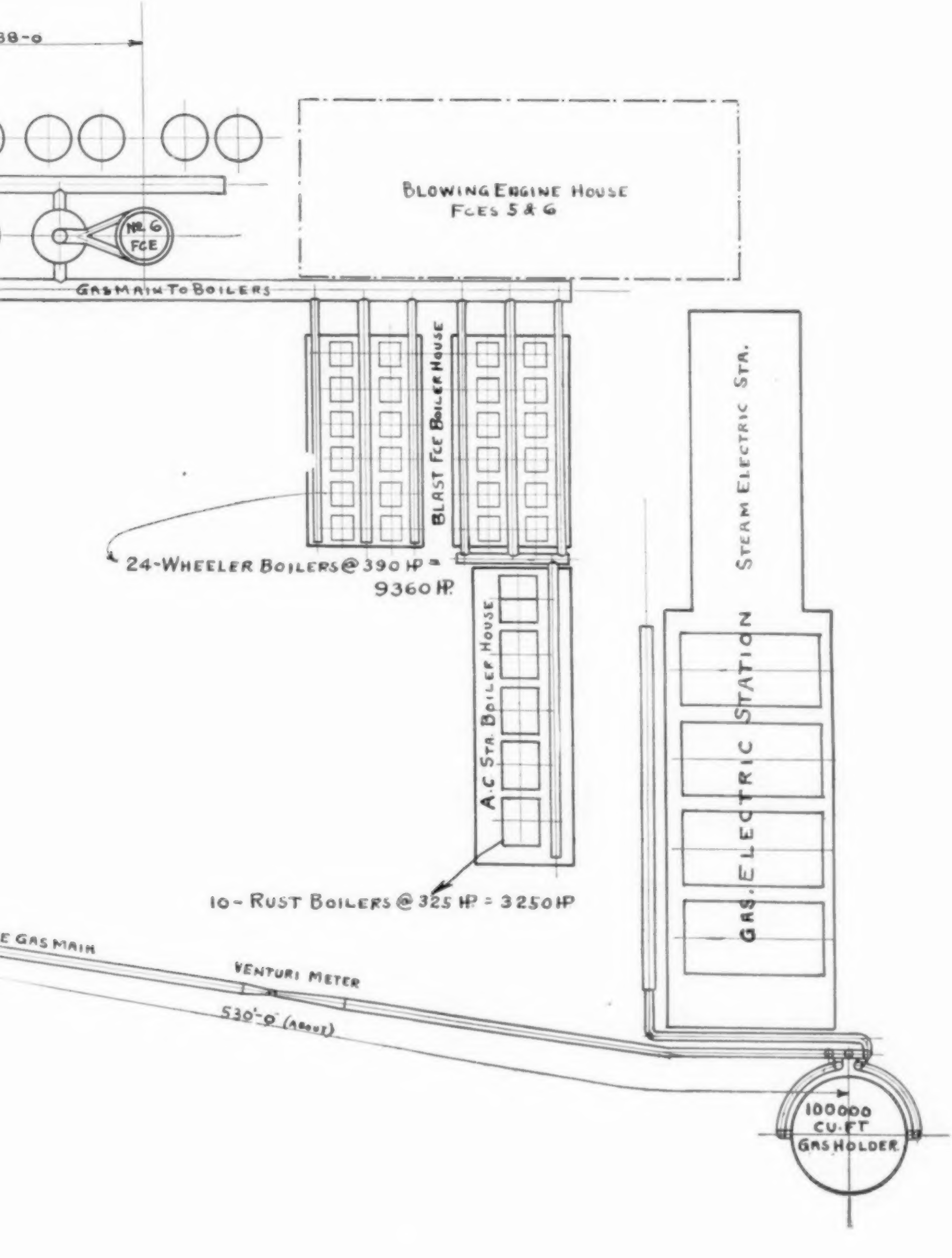


FIG. 3 ARRANGEMENT OF GAS POWER PLANT 1908


BLOWING ENGINE HOUSE
FCES 5 & 6
NO. 6
FCE
GAS MAIN TO BOILERS
BLAST FCE BOILER HOUSE
24-WHEELER BOILERS @ 390 HP =
9360 HP.
A.C. STA. BOILER HOUSE
10 - RUST BOILERS @ 325 HP = 3250 HP
VENTURI METER
530'-0" (ABOUT)
STEAM ELECTRIC STA.
GAS. ELECTRIC STATION
100000
CU. FT
GAS HOLDER

The average composition of the gas at the power station for the same period was:

CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.
12.6	26.64	3.2	2.33	96.73

The analyses of the gas of blast furnace No. 3 and the gas at the power house, coincide very closely, establishing proof that during this time this one furnace was furnishing the gas for the engines almost exclusively.

26 Since abandoning the boiler house for blast furnaces Nos. 1 to 4, which now receive steam through a 14-in. steam line from the boiler house at the east end, all gas from furnaces Nos. 1, 2 and 3 is being delivered to the gas-cleaning plant, with the exception of a small portion which goes to a small boiler house at the west end, while the gas of the remainder of the furnaces flows in the same direction to the boiler houses of furnaces Nos. 5 and 6 and the electric station. To illustrate the present condition of gas distribution, comparative data were compiled, given in Tables 3 and 4 in Appendix 2, which give the averages of gas analysis and heat values of the individual six furnaces, all of which were in blast in September 1909 and the average composition of mixtures of the gas from various furnaces calculated from the former. For the same month when these averages were taken the average composition and heat value of the gas delivered at the power house were:

CO_2	CO	H	CO_2	B.t.u.
10.03	29.80	3.77	2.98	108.70

Comparing this analysis with the mixture characteristics, given in Table 4, Appendix 2, the gas appears to be most nearly equivalent to the mixture from furnaces Nos. 1, 2 and 3 together.

27 Conditions were decidedly better in the second half of 1909, so far as uniformity of the gas supplied to the engines is concerned, but it is easily seen that changes in furnace operation must even under present conditions affect the quality of the engine gas. Whenever the gas supply from the furnaces on which the gas-cleaning plant is directly drawing happens to cease, in other words during checks or repairs, or if one or several of these furnaces are in trouble, disturbances are created in the regular flow and therefore in the quality of the gas, so that momentarily, or for longer periods, richer or leaner gas from other furnaces near the gas-cleaning plant intake is delivered to the gas engines. That such disturbances exist was very strikingly

proved in many instances. The gas engines, which had been operating smoothly, apparently receiving very uniform gas, would suddenly begin to backfire, or to have premature explosions and become very unsteady. These abnormal occurrences would be repeated at short intervals, although possibly lasting only a few minutes. The operating engineers soon discovered the cause of their troubles, and reported in their language that "a bad batch of gas" had caused the backfiring or the premature explosions and the "swinging" on the line. Such pronounced "waves" in the quality of the gas will often affect first the engine nearest to the gas holder, the trouble gradually extending to the engines down the line, and will stop first at the engine where the trouble started, gradually lessening on the rest of the engines, or else all engines will be affected simultaneously.

28 These interesting phenomena, and their bad effect on the parallel operation of the gas engines, prompted investigations which almost invariably located the causes for the sudden increase in hydrogen and methane. It was found that slipping of the furnaces was very frequently followed by backfires and premature explosions; and whether or not part of the raw wet stock in the furnaces reaches the incandescent zone, due to the upheaval of the material inside the furnace during slipping, thereby causing the formation of excessive amounts of hydrogen, remains an open question. Violent premature explosions and backfiring could in very many cases also be traced back to leaking tuyeres or hot blast valves, and these were so pronounced at times that the gas engines often fairly served as an indicator of such leaks.

29 The following gas analyses made on February 10, 1909, at the power house, give a good illustration of the suddenness of these changes:

GAS ANALYSES AT POWER STATION LABORATORY

ENGINE GAS

Time	CO_2	CO	H	CH_4	B.t.u. by analysis
11.00 a.m.	14.9	24.5	3.5	0.2	90.9
12.30 p.m.	13.8	24.7	4.3	0.3	94.6
3.30 p.m.	14.5	25.0	6.5	0.1	99.9
4.10 p.m.	14.2	24.8	4.5	0.2	94.7

The increase in hydrogen of almost 100 per cent between the first and third analyses is noteworthy. The effect on the engines was the occurrence of violent premature explosions around 3 o'clock of that

day. Backfiring happened simultaneously on three engines in operation on August 17, 1908, and was caused by fluctuations in the composition of gas. The daily chemical report for 24 hourse nding 6.00 a.m., August 18, contains the following record:

CHEMICAL ANALYSIS, AUGUST 18

Time	CO_2	CO	H	CH_4	B.t.u. by analysis
9.30 a.m...........	10.9	27.6	3.6	0.2	101.2
1.30 p.m...........	6.4	33.2	4.4	0.2	121.6
2.00 p.m...........	8.4	30.2	4.4	0.2	111.9
4.00 p.m...........	8.0	30.2	3.5	0.2	109.4

30 The simultaneous calorimeter determinations gave the following heat values:

CHANGE IN HEAT VALUE BY CALORIMETER

Time	B.t.u.	Time	B.t.u.
9.00 a.m......................	101.3	1.00 p.m......................	115.2
9.30 a.m......................	102.7	1.30 p.m......................	118.3
10.00 a.m......................	103.9	2.00 p.m......................	110.7
10.30 a.m......................	105.0	2.30 p.m......................	110.3
11.00 a.m......................	105.9	3.00 p.m......................	112.0
11.30 a.m......................	106.4	3.30 p.m......................	108.5
12.00 m......................	108.6	4.00 p.m......................	109.3
12.30 p.m......................	109.5	4.30 p.m......................	110.4

The heat value of the gas increased almost 20 B.t.u., or about 20 per cent in less than three hours, due to heavy coke blanks charged at blast furnace No. 1 which was in trouble. In this particular instance it was the sudden increase in carbon monoxid which caused the backfiring. It was not always possible, however, to prove by analysis or by calorimeter test that a sudden change in the heat value of the gas or a momentary increase in hydrogen had taken place when premature firing occurred; nevertheless, following the example of the Lackawanna Steel Company, the pressure of the cooling water for tuyeres and hot blast valves was reduced to a little below the normal blast pressure on all furnaces. Thus water-leaks into the furnaces were very effectively stopped and one principal cause for premature explosions at the engines was removed.

31 The kind of iron produced by the different furnaces at different times had a considerable effect upon the quality of the gas. Thus in September 1909 the average heat value of the gas at the power station was 108.7 B.t.u. per cu. ft., because during this month blast furnace

No. 1 was making ferrosilicon with a coke consumption of over 4600 lb. per ton of product. The average analysis of the gas of this furnace is given in Table 3, Appendix 2, and the composition of the engine gas for September is given in Par. 26 of the paper. The richest gas which the engines ever received occurred September 17, 1909, the average of the analyses for the day being as follows:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
4.7	34.9	3.2	0.16	7.92	123.3

The average corresponding B.t.u. values determined by calorimeter were 122.5 per cu. ft. The influence of this rich gas on the operation of the engines is shown in the daily record of engine operations for the same day. Three engines were running and were backfiring and hav-

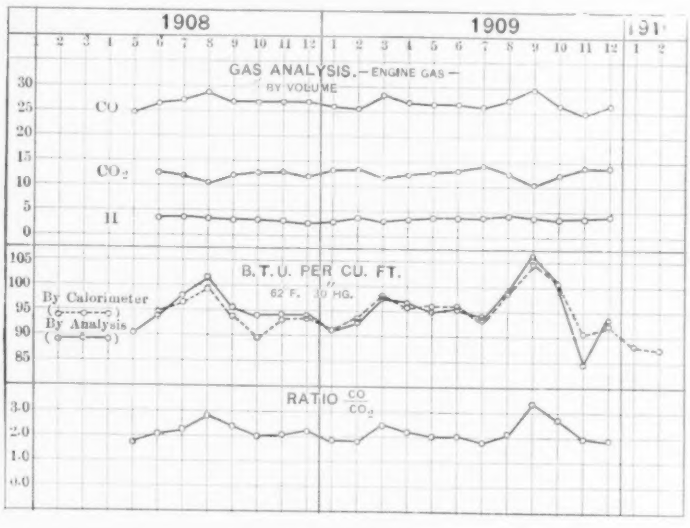


FIG. 4 GAS ANALYSIS AND HEAT VALUE OF BLAST FURNACE GAS (MONTHLY AVERAGES)

ing premature explosions all day long. On the following day the operating engineer reported that the gas made a quick change about 1.00 a.m. to very poor quality, causing all engines to misfire and to drop about one-half of their load, and the richness of mixture had to be changed on all engines to obtain proper ignition. About 2.00 a.m. the gas became suddenly very rich and the engines again backfired heavily, necessitating additional changes in the mixture. The leanest gas

on which these engines were operated during the year 1909 occurred in November, the average composition for the month being as follows:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u.
13.8	24.7	3.59	0.19	1.79	86.7

The lowest daily average heat value occurred on November 16, with gas of the following analysis:

CO	CO_2	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
15.8	22.1	3.3	0.3	1.39	83.1

The gas was so poor that day that it was impossible to keep it burning in the calorimeter. The lowest heat value ever recorded is 79.5 B.t.u. per cu. ft. on November 17, the gas analysis at 12.00 o'clock noon giving the following results:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
17.1	21.6	3.1	0.1	1.26	79.5

The effect of such very poor gas on the engines is quite noticeable; the full output of the generators could not be maintained, although the proportion of air and gas was changed to meet the new conditions.

32 In Appendix 2, Table 5, is given the average composition of the blast furnace gas for each month, averages for the first and second halves, and the average for the whole year of 1909; further, the heat value of the gas determined by calorimeter. In Fig. 4 herewith these values are plotted for each month since June 1908 when the systematic records were begun. The discrepancies in the heat values as computed, and as determined by Junkers calorimeter, are explained by the fact that analyses are made about every three hours, while calorimeter readings are taken almost continuously during the day. The number of observations is therefore much greater for the latter than for the former. For all calculations the heat values determined by calorimeter are used exclusively. The methods of gas sampling and analysis used as described by Mr. L. A. Touzalin are as follows:

Daily samples of blast furnace gas are taken at each individual furnace, all samples being accumulative and representing a fair average of the gas production extending over a period of 24 hours. The sample is taken between down comerstand pipe and dust catcher (as shown in Fig. 5) and conducted by means of a 2½-in. or 2-in. pipe, first to a miniature dust catcher and then to two washing bottles connected in series by means of a 2-in. pipe. From the second bottle

the gas passes to a sampling tank of 5 cu. ft. capacity, made of galvanized iron and of regular gas-holder construction. All water used in the washing bottles and in the sampling tank is first saturated with blast furnace gas. The small dust catcher consists of an 18-in. length of 6-in. iron pipe capped at each end and suspended in a vertical position. By removing the cap at the bottom end, the accumulated dust may be cleaned out as often as necessary. The valve placed at the lower end permits a small stream of gas to flow continuously through the sampling pipe into the small dust catcher and escape into the air. A great deal of dust escapes with this gas, and besides preventing continuous clogging, this valve allows any condensed water to drain off. The screw cock attached to the rubber tube between the washing bottles affords means of so adjusting the rate of flow that the tank is almost filled in 24 hours. At the end of each period a sample of the accumulated gas in the tank is withdrawn into a glass gas holder of 250 cu. cm. capacity, while the remaining gas in the tank is allowed to escape into the air. The bell of the tank thus drops down in place and the gas is again started for the next 24-hour sample. The 250 cu. cm. sample of the gas in the gas holder is taken to the laboratory for analysis. At the power station, where a special gas laboratory, fully equipped, is installed, gas samples are taken directly from the gas main between gas holder and engines. Gas analyses are made several times during the day and as often as necessary if any unusual occurrences at the engines indicate a change in gas quality.

33 Daily gas analyses and heat values as well as results of dust and moisture determination, together with additional chemical information, are recorded on daily report sheets (Fig. 6).

DESCRIPTION OF GAS-CLEANING PLANT

34 When the installation of blast furnace gas engines was decided on in 1906, very little information and experience on the important matter of gas cleaning was available in this country. Some experiments had previously been made at the plant under discussion on a small scale, with different designs of wet scrubbers and baffle washers, the deception generally prevailing at that time that gas could be cleaned sufficiently for engine purposes by so-called "static" methods, that is, by passing it through towers filled with baffle plates and a checker work of wood or iron, and sprayed with water in finely divided form. The results of these experiments were discouraging, as might have been expected, and the installation of Theisen gas washers for refining the gas was eventually decided upon.

35 The blast furnace gas for the gas engines is cleaned in two distinct stages. It is first subjected to a preliminary dry and wet scrubbing in the so-called primary or preliminary cleaning plant, and subsequently undergoes a secondary cleaning or refining by Theisen washers in the secondary washing plant. Fig. 3 shows dia-

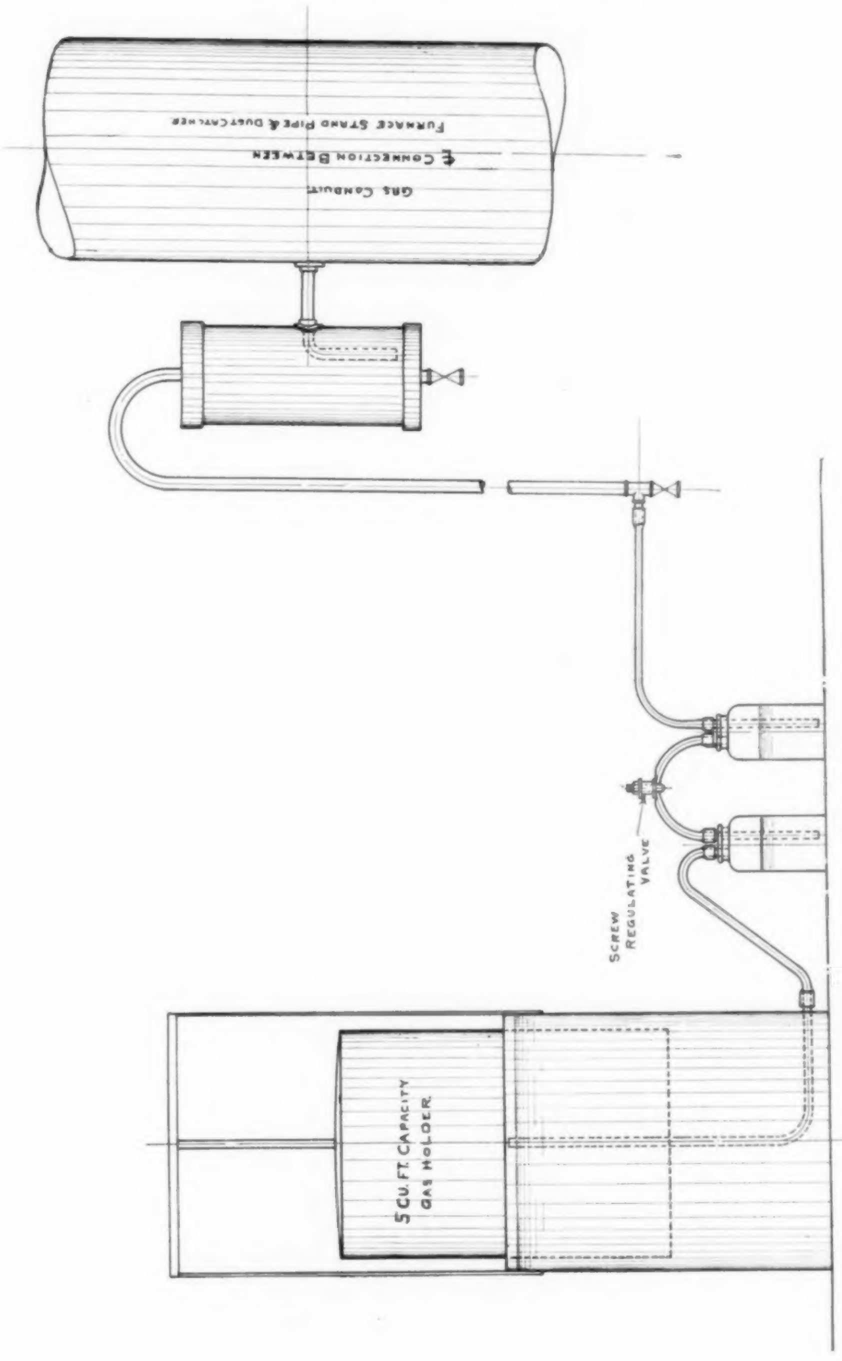


FIG. 5 APPARATUS FOR CONTINUOUS GAS SAMPLING

DAILY CHEMICAL REPORT — For 24 Hours Ending 6 A.M. Tuesday, Sept. 21st [illegible]

BLAST FURNACE GAS — Samples taken at Gas power station laboratory

TIME	CHEMICAL ANALYSIS % By Volume						B.T.U. Per Cubic Foot	
	CO 2	CO	H 2	CH 4	$\frac{CO_2}{CO}$	$\frac{CO}{CO_2}$	By Analysis	By Junkers' Calorimeter
9:00 A.M.	8.1	31.1	3.6	.2	.260	3.84	112.6	112.0
9:30								111.9
10:00								111.0
10:30								121.2
11:00	8.2	32.2	3.7	.2	.255	3.92	116.5	119.9
11:30								116.2
12:00 NOON	8.2	32.2	3.0	.1	.255	3.92	113.6	114.0
12:30 P.M.								113.1
1:00								112.8
1:30								111.1
2:00								110.7
2:30								113.1
3:00								112.5
3:30								112.7
4:00	8.7	31.0	5.0	.1	.280	3.56	115.2	114.3
4:30								114.7
AVERAGE	8.3	31.6	3.8	.2	.262	3.81	114.5	113.8

		DAY TURN	NIGHT TURN	AVERAGE
Weather		Fair	Clear	—
Barometer		29.40″	29.31″	29.35″
Direction of Wind		Southwest		
Test Started		8:40 A.M.	5:00 P.M.	—
Test Ended		4:40 P.M.	8:00 A.M.	—
Number of cubic feet of gas passed through filter	Before Scrubbers	39.10	54.20	46.65
	After Scrubbers	59.22	100.63	79.92
	After Theisens	—	—	—
	A.C. Station Gas	88.36	148.37	118.37
	Blowing Eng. Gas	—	—	—
Moisture Grains per cu. foot	Before Scrubbers	—	—	—
	After Scrubbers	14.719	—	—
	After Theisens	—	—	—
	A.C. Station Gas	8.984	—	—
	Blowing Eng. Gas	—	—	—
	In Air	6.49	6.15	6.32
Flue Dust Grains per cu. foot	Before Scrubbers	3.8830	4.0812	3.9821
	After Scrubbers	.8147	.6884	.7516
	After Theisens	—	—	—
	In A.C. Engine Gas	.0085	.0052	.0068
	In Blowing Eng. Gas	—	—	—
% Dust Removed-Scrubbers		79.0	83.1	81.0
% Dust Removed-Theisens		98.9	99.2	99.0

Note-All figures given on this report are reduced to standard conditions i.e.
Temperature 62°F, Barometer 30″ Mercury
The heat value Constants are;
CO...324.45 B.T.U. PER CUBIC FOOT
H2...274.78 " " " "
CH4...915.08 " " " "

Metric Conversion Factor
1 Gram-15.43 Grains
MacFarland's tables of Reduction
Factors are used for all calculations within their scope.

OBSERVATIONS-
Furnace 1 on Ferrosilicon.
" 2 Basic iron.
" 3 " "
" 4 " "
" 5 Bessemer iron.
" 6 " "

3 Gas engines in operation.

Wm. Brady. CHIEF CHEMIST
Peterson.

FIG. 6 FORM FOR DAILY CHEMICAL REPORT

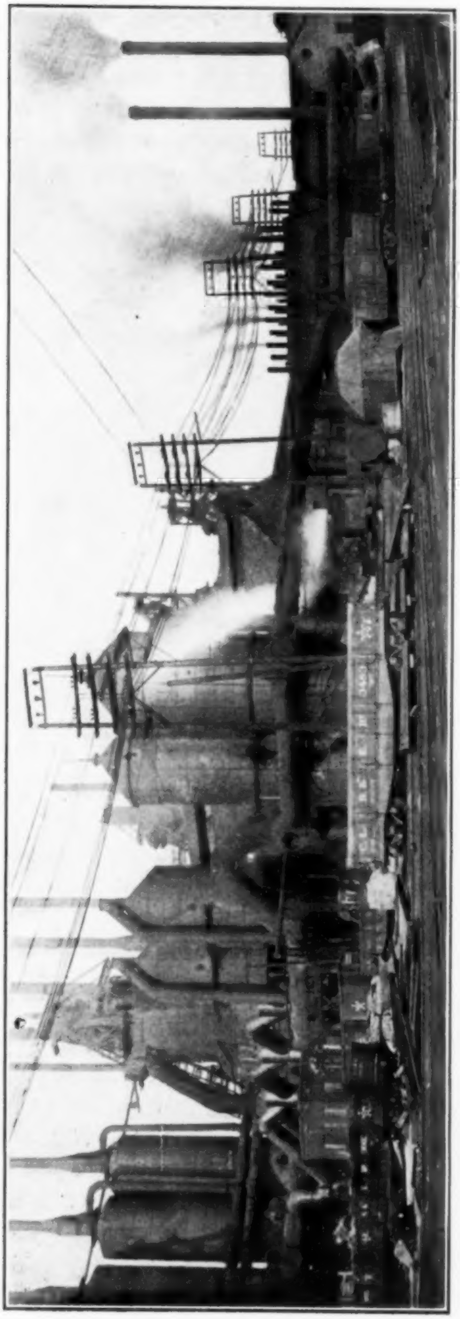


FIG. 7 GENERAL VIEW OF GAS-CLEANING PLANT, 1908

grammatically the general arrangement of the complete cleaning plant as it existed in 1908, while Fig. 7 gives a photographic view. When the gas-cleaning plant was designed in 1906, the raw gas was not cleaned except by the usual small dry dust catchers at the end of the downcomers of each furnace, and it was decided to install two special dry dust catchers of large capacity to remove the bulk of the heavy dust.

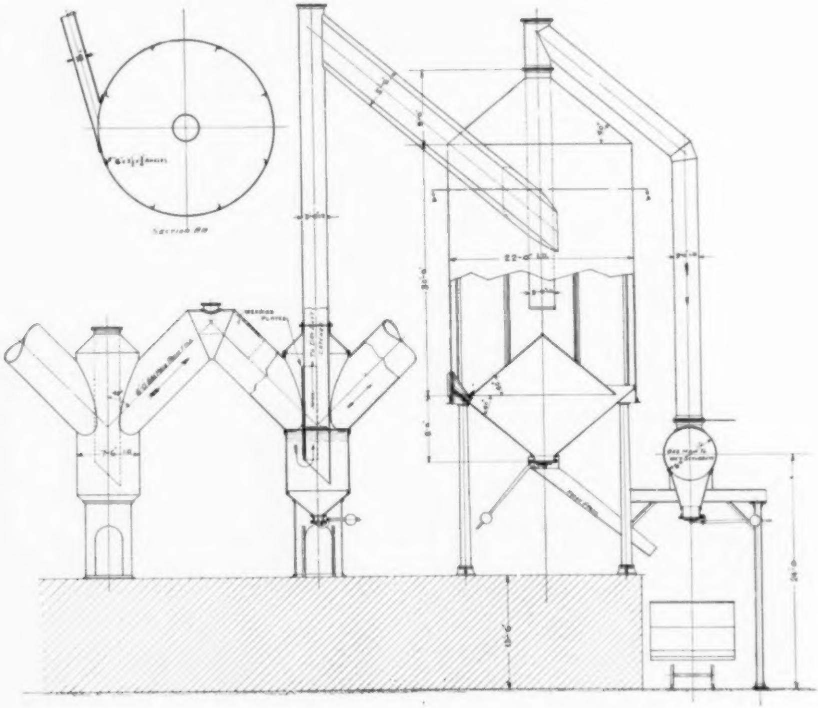


FIG. 8 DRY DUST CATCHER

PRELIMINARY GAS-CLEANING PLANT

36 The raw gas on leaving the overhead gas flue first passes a water seal serving to shut off the gas power plant from the general system in case of necessity, and enters an unlined self-cleaning zigzag gas flue 6 ft. in diameter. Fig. 8 shows in detail the plan, and Fig. 9 a photograph of the zigzag flue and the dry dust catchers. It was originally intended to increase the capacity of the preliminary cleaning plant subsequently, by the addition of enough dry and wet scrubbers to clean the total quantity of gas produced by all six furnaces, for use

under boilers and in hot blast stoves, and provision for these additions was made when designing the cleaning plant. By means of water seals in the "dust legs" supporting the zigzag flue, and spectacle valves at the points of discharge of the dry-dust catchers into the "collecting main," each dust catcher may be shut off during the operation of the plant, in case of repairs or cleaning. As seen in the illustrations these water seals were designed on the principle of the Crawford valve, and by cutting off the ends of the inside pipes at an angle, it was intended to provide means of regulating the amount of gas passing through each dry dust catcher by filling the seal with water to a certain height, which was regulated by telescopic overflow.

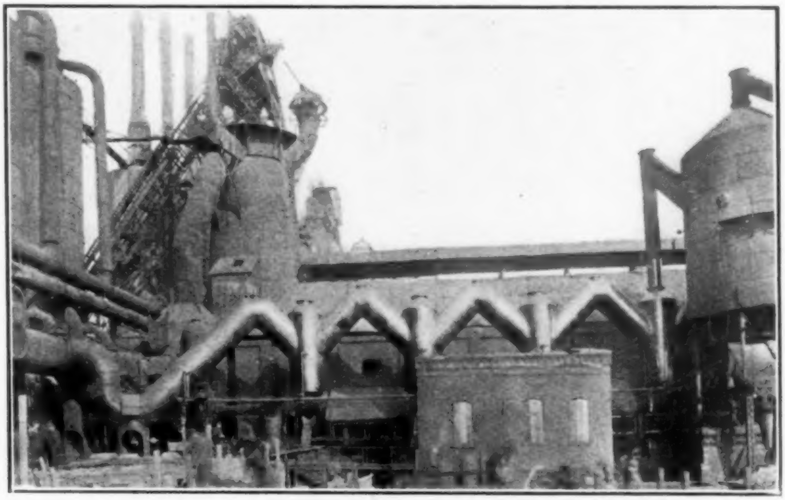


FIG. 9 ZIGZAG MAIN AND DRY DUST CATCHER

37 Neither zigzag main nor dry dust catchers were lined with fire brick, as had heretofore been the practice with all gas pipes conveying raw gas of high temperature, as it was desired to take advantage of the reduction of temperature by radiation of heat through the unlined plate work. The results given elsewhere prove that the desired object was very satisfactorily accomplished. At all points of sudden change in direction of the flow of gas, "wearing" plates were provided, as excessive wear of the plate work, from the impinging of the dust-laden gas, was expected. These plates can be removed and replaced by new ones, through manholes arranged for this purpose. The dust

legs, as well as the dry dust catchers, were raised above the yard level high enough to permit the disposition by gravity of the accumulated flue dust into railroad cars by means of bell valves and dust spouts.

38 The dry dust catchers, two in number and operating in parallel, were 22 ft. in diameter by 31 ft. high with 9-ft. cones at each end. The choice of the diameter of the dry dust catchers, as well as of the

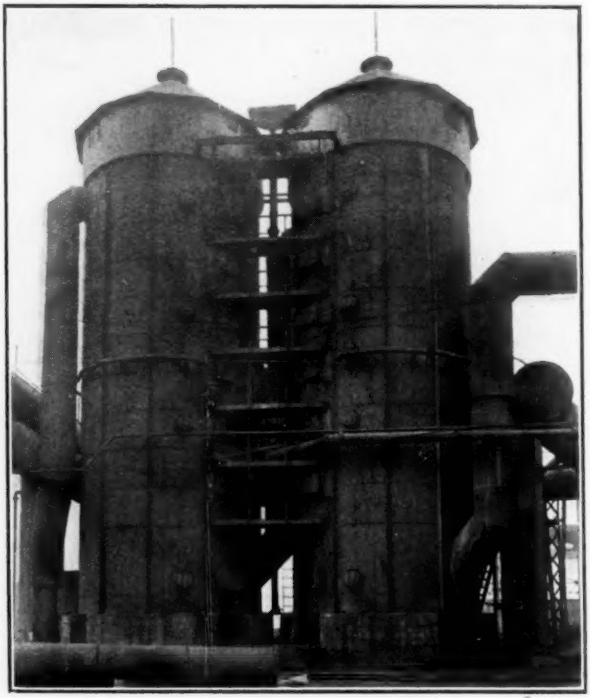


FIG. 10 WET SCRUBBERS

wet scrubbers, was accidental, as it happened that some old 22-ft. hot blast stove shells were available at the time. The dry dust catchers were given tangential gas inlets, assuming that by this arrangement, and by inclining the flattened gas-inlet pipe, the gas would be caused to travel in long spirals from top to bottom, thus lengthening the path of the gas, and angle irons were placed vertically on the inside of the shell to provide for increased friction while the gas was traveling through the dust catchers at the slow rate of 1.5 ft. per sec. The

bottom cone was separated from the cylindrical part of the dust catcher by an inverted cone arranged umbrella-wise to prevent the dust accumulated in the bottom cone from being stirred up by disturbances caused by furnace slipping and re-entering the gas. The gas left the dry dust catchers near the apex of the umbrella, passing through a self-cleaning pipe into the collecting main and hence to the wet scrubbers. Explosion doors were arranged for on the dry dust catchers, but were eventually bolted down as unnecessary.

39 From the 6-ft. 6-in. collecting main, the dry-cleaned gas passes to the wet scrubber, shown in Figs. 10, 11*a*, 11*b* and 11*c*, through self-cleaning pipe lines. The piping arrangement permits the operation of the original two wet scrubbers in series, or in parallel, by turning a num-

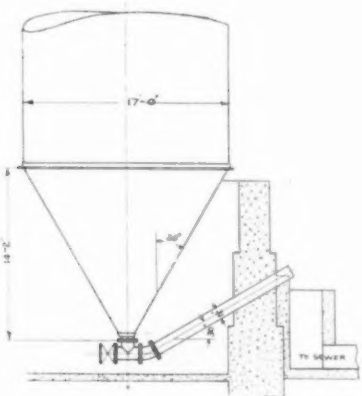


FIG. 11*a* DETAIL OF SCRUBBERS NOS. 1 AND 2

ber of spectacle valves, and water seals allow the shutting-off of either wet scrubber for cleaning, without interfering with the operation of the power plant. From the start the two wet scrubbers were operated in series, the total quantity of gas consumed by the engines passing first one and then the other. The gas enters each wet scrubber at the bottom of the shell, which is 22 ft. in diameter and 55 ft. in height. The inside is divided horizontally into six compartments, each containing eight rows of slats made of clear No. 1 white pine dressed all over. Each system of slats is supported independently by I-beams and angle irons riveted to the shell. The slats are 5 in. high and $\frac{7}{8}$ in. thick, by about 5 ft. 6 in. long, and ten slats on an average are nailed to distance pieces, forming hurdles about 5 ft. 6 in. long, and 3 ft. 7 in. wide, a size and weight which permit of easy handling. An

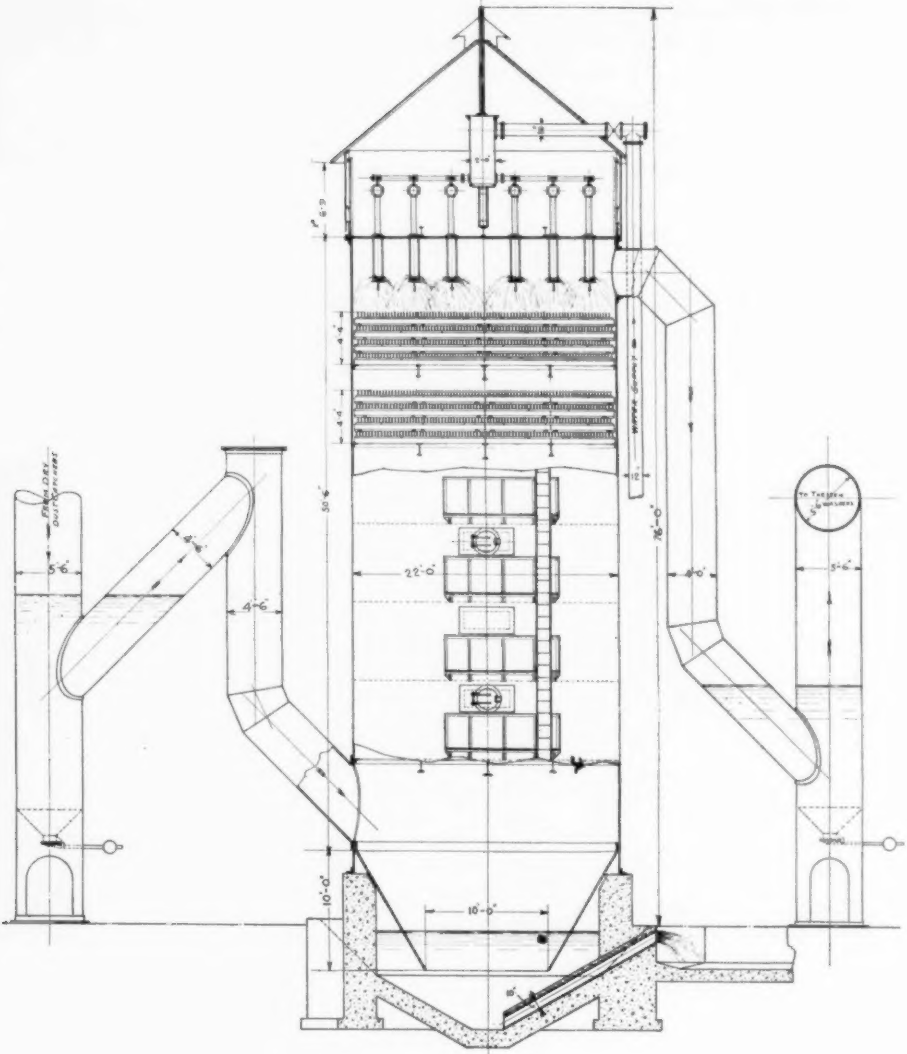


Fig. 11*b* Section of Scrubbers Nos. 3 and 4

interior view of the scrubbers with the hurdle arrangement is shown in Fig. 12.

40 Profiting by the experience gained elsewhere with wet scrubbers of the same kind, flue dust bridging over between slats and clogging the hurdles, it was decided to space the slats in the following manner;

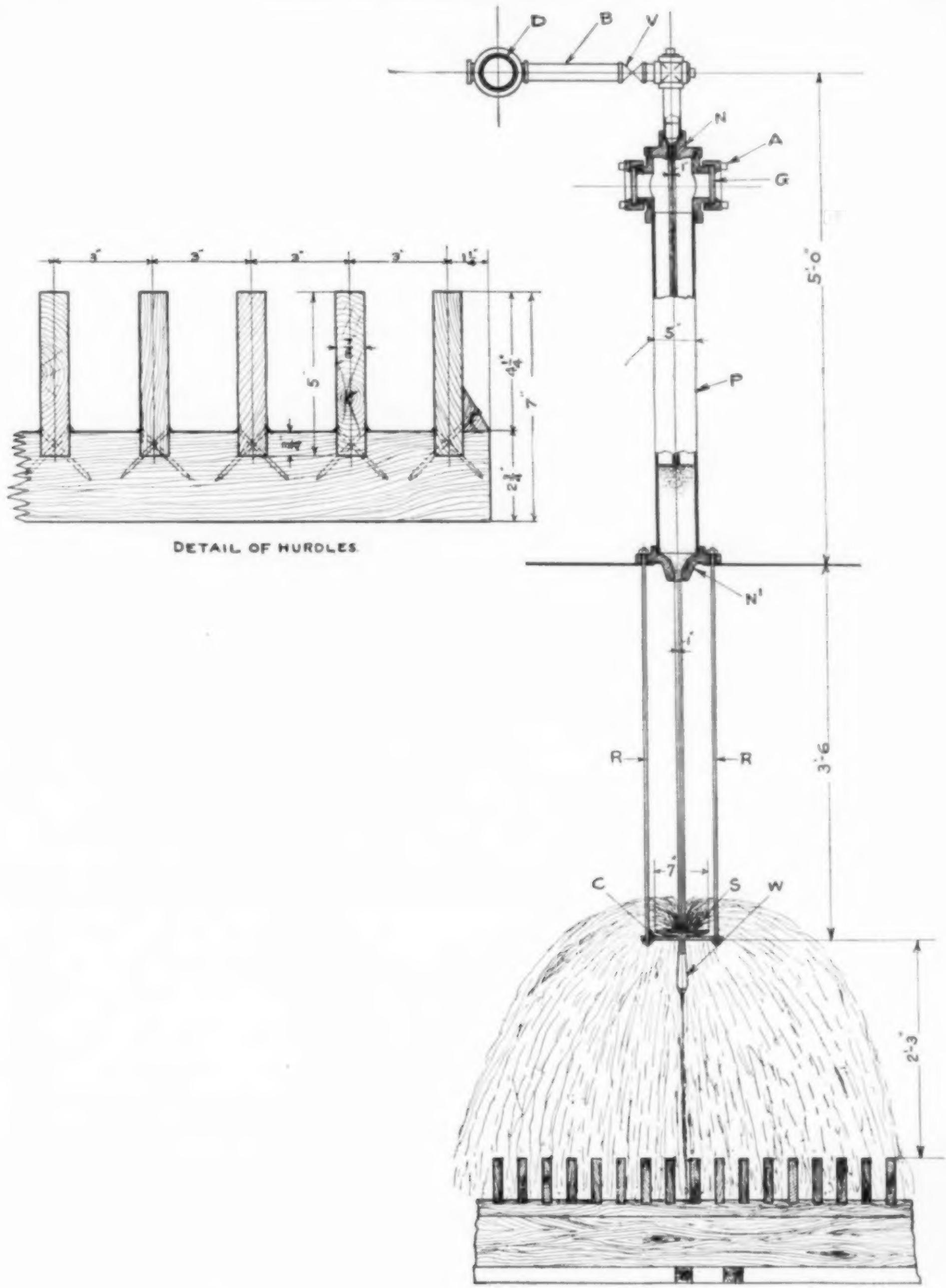


FIG. 11c DETAIL OF SPRINKLERS AND HURDLES

In the three lower compartments of scrubber No. 1, which receives dry-cleaned gas containing a large amount of coarse flue dust, the average distance of the slats was made 9 in. while in the three upper compartments the slats have 6 in. spacing. In the three lower compartments of scrubber No. 2, receiving gas already washed in the first scrubber, the slats were spaced with 4½ in. centers, while the corresponding distance is 3 in. in the three upper compartments of the second scrubber. All hurdles were placed in the different rows and compartments in such a way that the slats of each upper row straddle the slats of the row immediately below, thus obtaining a continuous

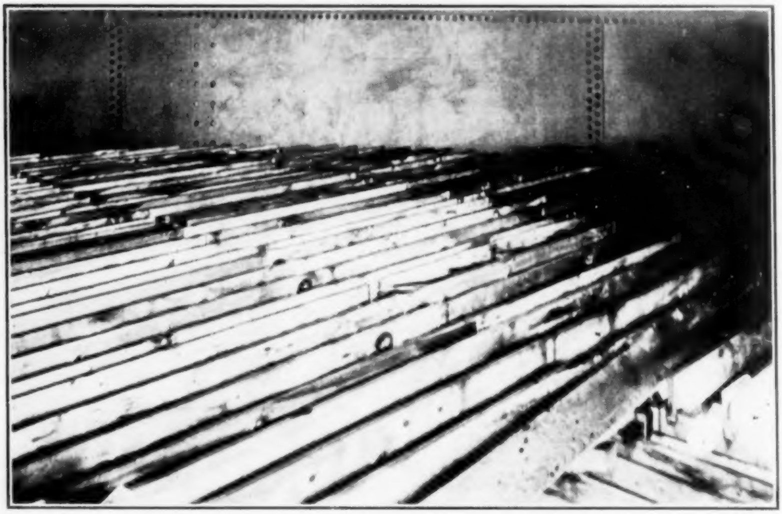


Fig. 12 Interior View of Scrubbers Showing Hurdles

checker arrangement without any channels. A space of about two feet was left between each two hurdle sets, making each compartment accessible for cleaning or changing hurdles without removing the whole filling, and manholes and platforms were provided to facilitate this work.

41 The top of the wet scrubbers is formed by $\frac{5}{16}$ in. flat covers, supported by 8-in. I-beams. Each cover plate supports 36 sprinklers, shown in Fig. 13 and in detail in Fig. 11*c*, distributed over the entire section. Each sprinkler consists of two nozzles N and N', and a cast-iron spray plate S with slightly curved surface and weight W

to insure horizontal position. The spray plate is inserted in the center hole of the crosspiece *C*, which is supported by two rods *R* fastened to the top cover plate. The upper nozzle of 1 in. diameter is mounted on one branch of a cast-iron cross, while the opposite branch is connected by a 5-in. wrought-iron pipe *P* about three feet in length, to the lower nozzle *N′* of equal size. The other two branches of each cross are closed by caps *A*, containing plate glass discs *G*, $\frac{1}{4}$ in. thick and 4 in. in diameter. The open water tank located above the wet scrubber supplies the washing water under a small but constant head, through distributing pipes *D* and branch connections *B* to the sprinklers, the amount of water being regulated by valves *V*. The operation of these sprinklers is obvious. A stream of water falls in each sprinkler through a distance of about eight feet, breaking up into an exceedingly fine mist by impinging on the spray plates, and as the sprays of the 36 sprinklers overlap each other, the distribution of water is perfect.

42 These sprinklers were tested before the wet scrubbers were put into operation, and one minute after turning on the water there was not a dry spot on the inside of the scrubbers. Originally these sprinklers were without the nozzle *N′*. The action was the same as far as the distribution and the atomization of the water was concerned, but with the serious drawback that the dirty gas could reach the upper part of the sprinklers and deposit dirt on the sight glasses, which soon became useless. By the insertion of nozzle *N′* this trouble was successfully overcome, as the water flowing through this nozzle completely seals the upper part of the sprinklers so that the glasses can be removed during operation without danger from escaping gas. The great advantages of this type of sprinkler are that a clogging of the water passages can never occur, and uniform distribution of the water can always be obtained. Their operation has been exceedingly satisfactory. All water piping on top of the wet scrubbers is housed-in for protection against frost.

43 The lower part of the wet scrubbers dips with a conical extension into a water seal provided in the concrete foundation. The muddy water is carried off through an overflow pipe reaching to the bottom of the seal, thus keeping the water in constant circulation and thereby effectively preventing any accumulation of mud in the seal. It was found advantageous, however, to introduce into this overflow passage and reaching to the bottom of the seal, a 1-in. pipe through which water under pressure is constantly discharged, stirring up the

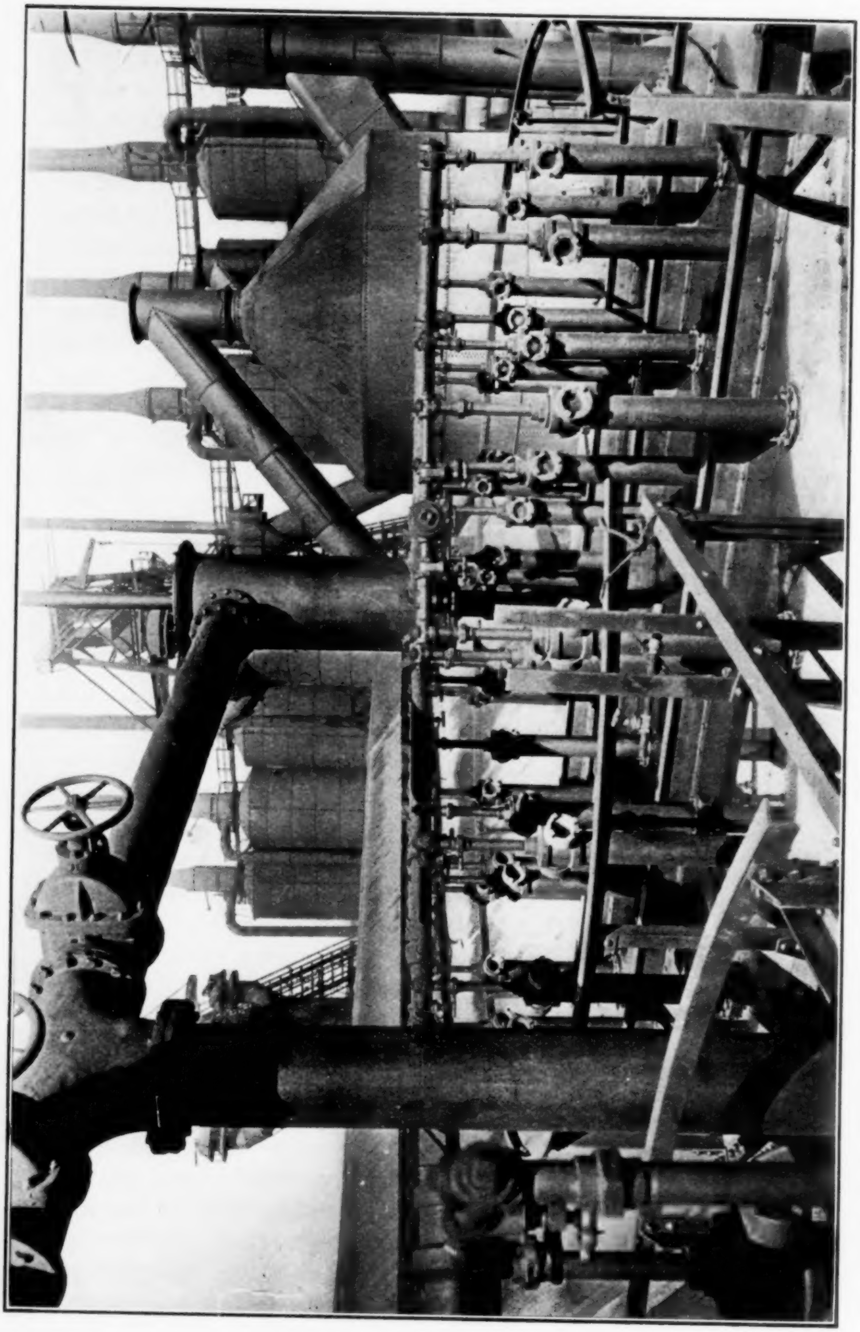


FIG. 13 WATER SPRINKLERS

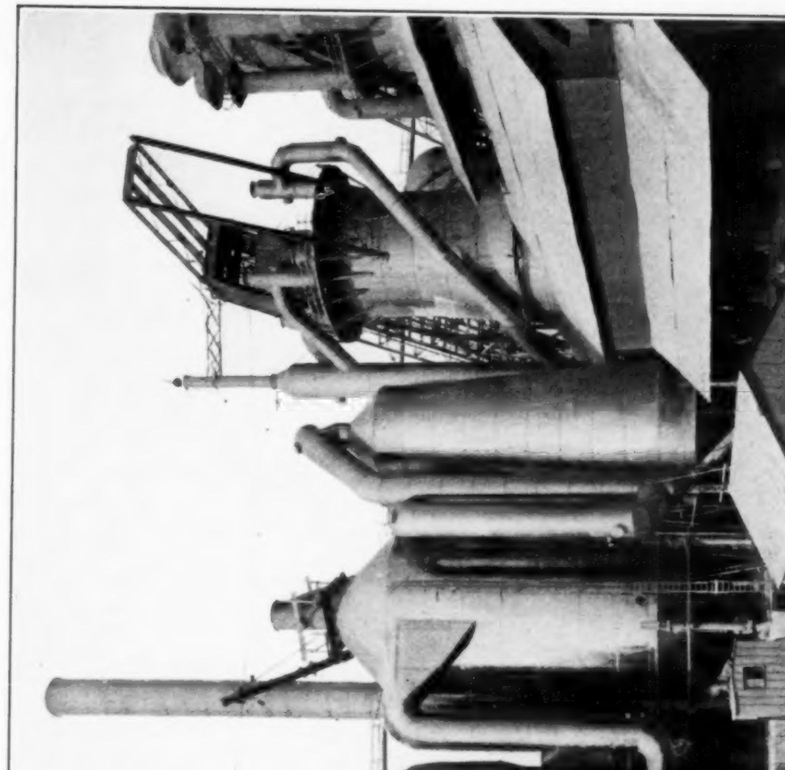


Fig. 14 Dry Dust-Catcher System at Furnaces 5 and 6

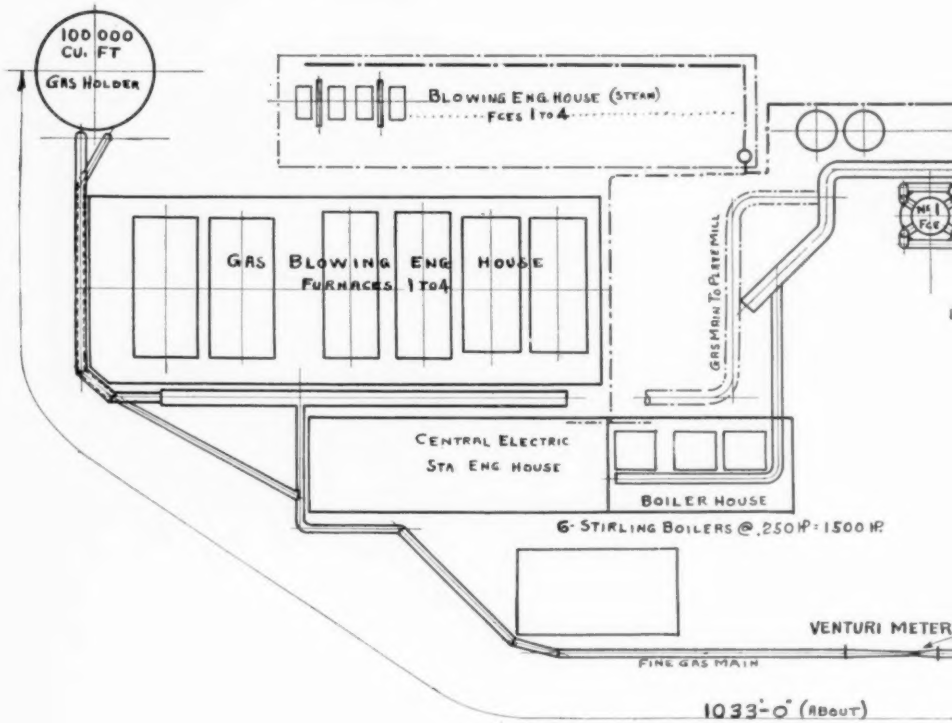
sediment and keeping the water in motion. On the two new wet scrubbers recently installed this type of water seal was abandoned in favor of the simple arrangement shown in Fig. 11*b*.

44 The waste water from these scrubbers flows into a large settling tank, which was installed to prevent a possible clogging of the main sewer. Both compartments of the settling tank, which was in operation about two years, have since been completely filled with mud, but as the dust in suspension does not prove to be troublesome in the sewers, little attention is being paid to their regular cleaning.

45 When in the early part of 1909 four additional gas blowing-engines for blast furnaces No. 1 to 4 were purchased, an increase in the capacity of the gas-cleaning plant was necessary, and since in the meantime the question of dry cleaning the raw gas at the furnaces had been solved, it was decided to change the two original dry dust catchers into wet scrubbers. As soon as the voluminous dry dust catcher system at the furnaces (Fig. 14) was in operation its effect was noticed in the materially reduced efficiency of the dry dust catchers in the preliminary gas-cleaning plant. These had formerly removed a great deal of heavy, dry flue dust, but suddenly became practically useless, and only a little dust, now in the form of mud, was taken out. While the change from dry dust catchers to wet scrubbers was being made, in the second half of 1909, only the two original wet scrubbers were in operation. In the near future four wet scrubbers, of sufficient capacity to take care of the preliminary washing of the gas required by 40,000 h.p. in gas engines, will be in use. Fig. 15 shows the general arrangement of the gas power installations at present. Fig. 16 is a diagram of the path of the gas through the new wet scrubbing plant, showing the combinations in which the four scrubbers can be operated. Fig. 7 shows the gas main carrying the clean gas from the wet scrubbers to the secondary cleaning plant. Attention is called to the design of the supports of this pipe line, which are built as so-called dust legs, wherein water and flue dust are deposited, and drawn off occasionally through bell valves. The clean gas main, while not self-cleaning, is arranged to slope in both directions. At certain intervals circular water pipes with spray arrangements are installed for flushing the clean gas main, and at the points where a sudden change of direction of the gas occurs sealed holes are provided, through which a thorough cleaning of the pipe line can be made, with fire hose and hign-pressure water, during the operation of the plant.

46 After leaving the wet scrubbers, the clean gas, as it is called, is in such a condition that it could be used under boilers and in hot blast

FOLDER No. 2


100 000
CU. FT
GAS HOLDER
BLOWING ENG HOUSE (STEAM)
FCES 1 TO 4
GAS BLOWING ENG HOUSE
FURNACES 1 TO 4
GAS MAIN TO PLATE MILL
CENTRAL ELECTRIC
STA ENG HOUSE
BOILER HOUSE
6-STIRLING BOILERS @ 250 HP = 1500 HP
VENTURI METER
FINE GAS MAIN
1033'-0" (ABOUT)

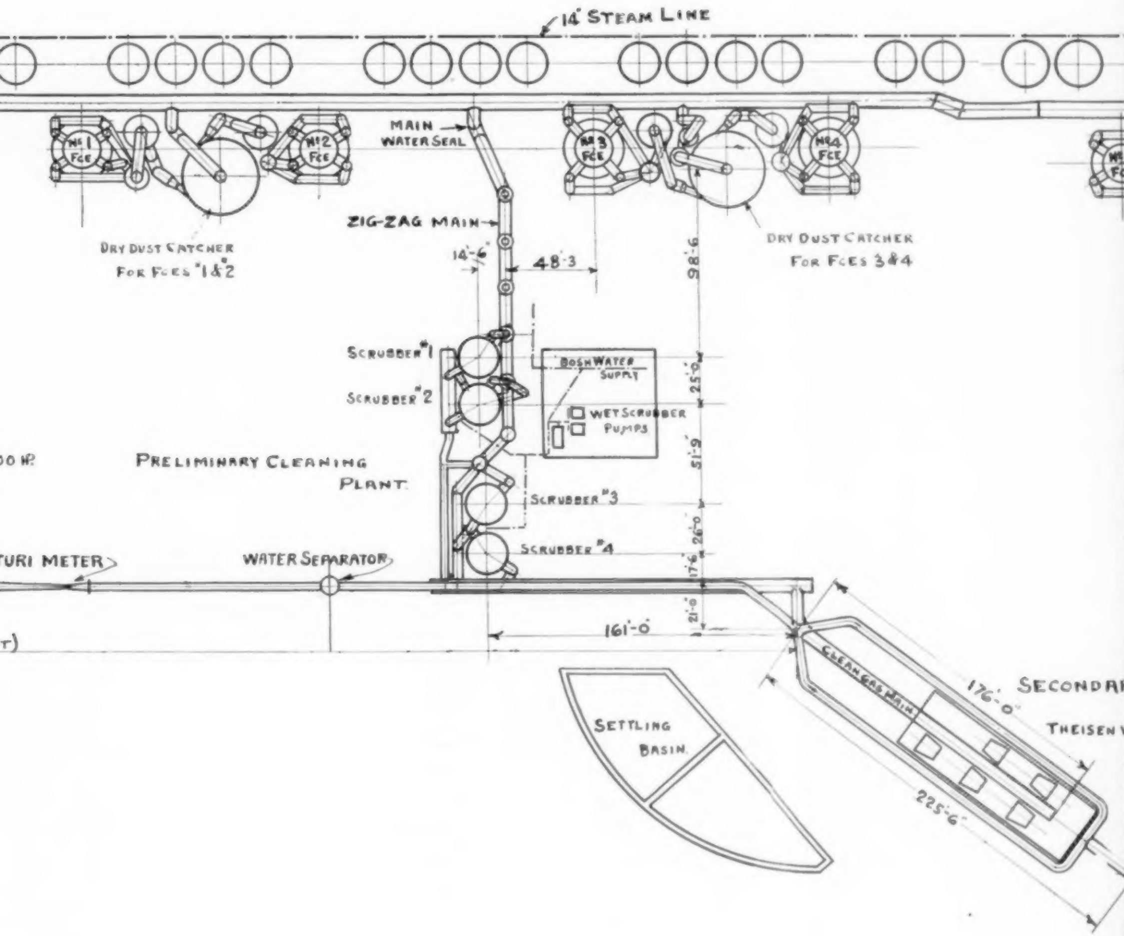


Fig. 15 Arrangement of Gas Power Plant 1910

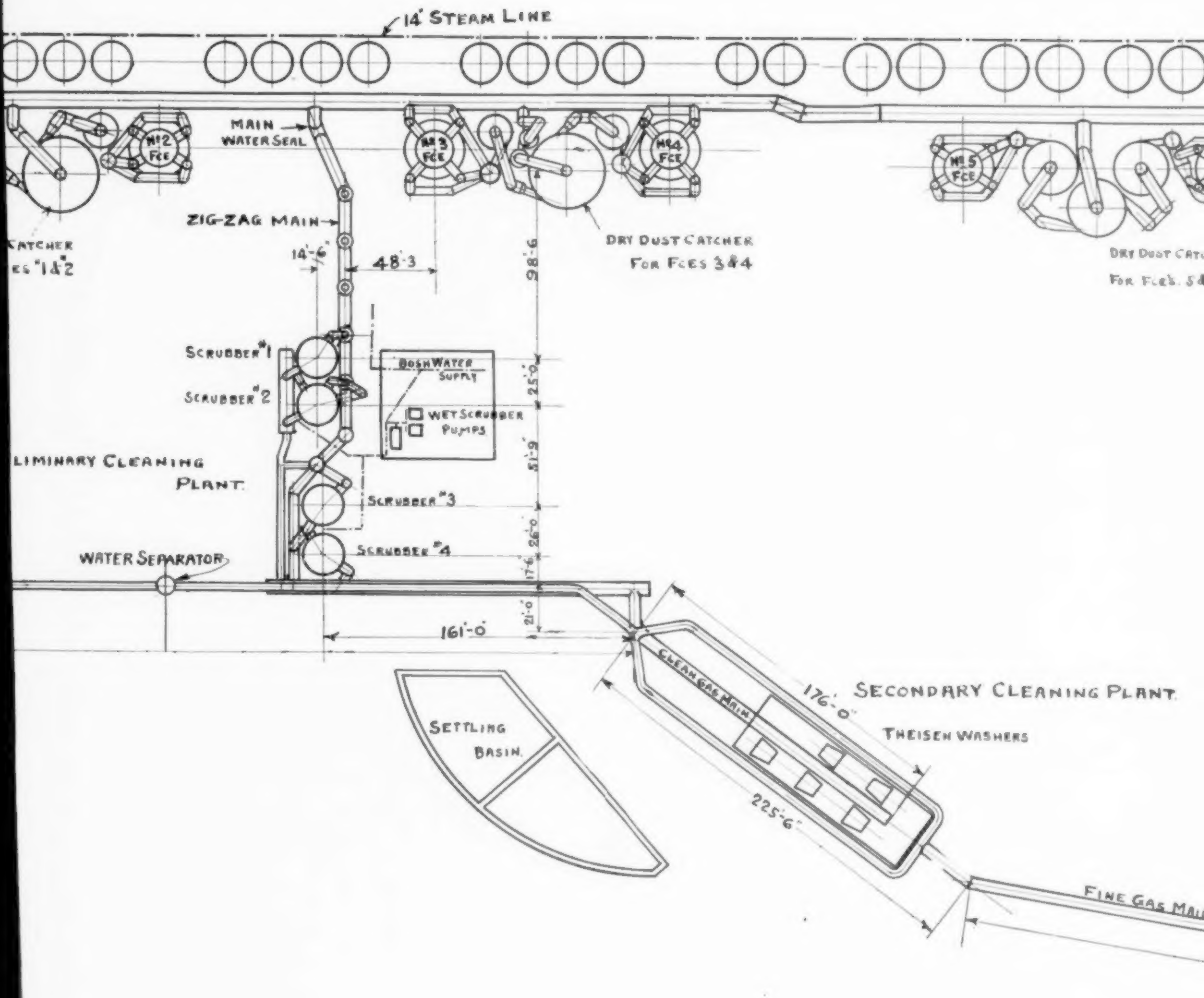
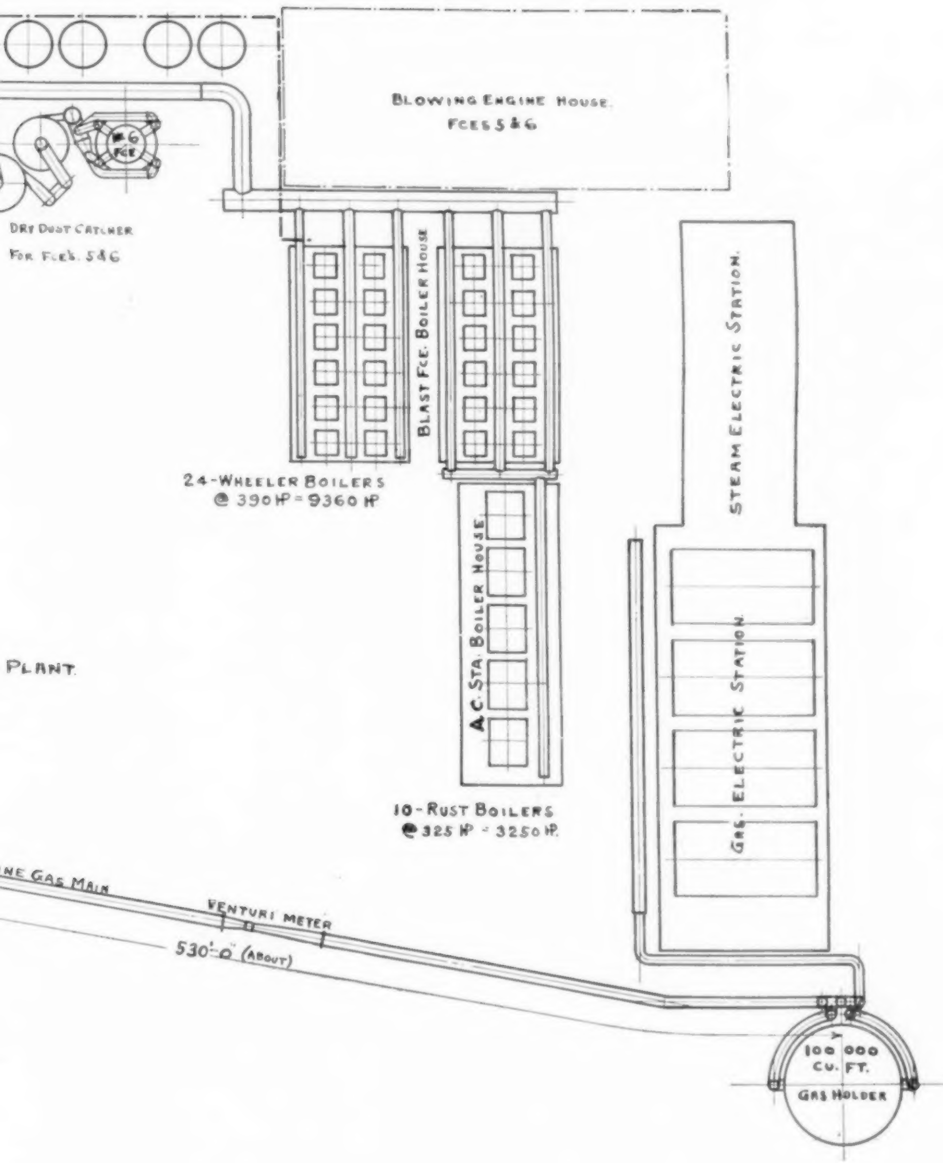


Fig. 15 Arrangement of Gas Power Plant 1910


BLOWING ENGINE HOUSE.
FCES 5 & 6
No. 6 FCE.
DRY DUST CATCHER
FOR FCES. 5 & 6
BLAST FCE. BOILER HOUSE
24-WHEELER BOILERS
@ 390 HP = 9360 HP
STEAM ELECTRIC STATION.
A.C. STA. BOILER HOUSE
PLANT.
10-RUST BOILERS
@ 325 HP = 3250 HP.
GAS. ELECTRIC STATION.
NE GAS MAIN
VENTURI METER
530'-0" (ABOUT)
100 000
CU. FT.
GAS HOLDER

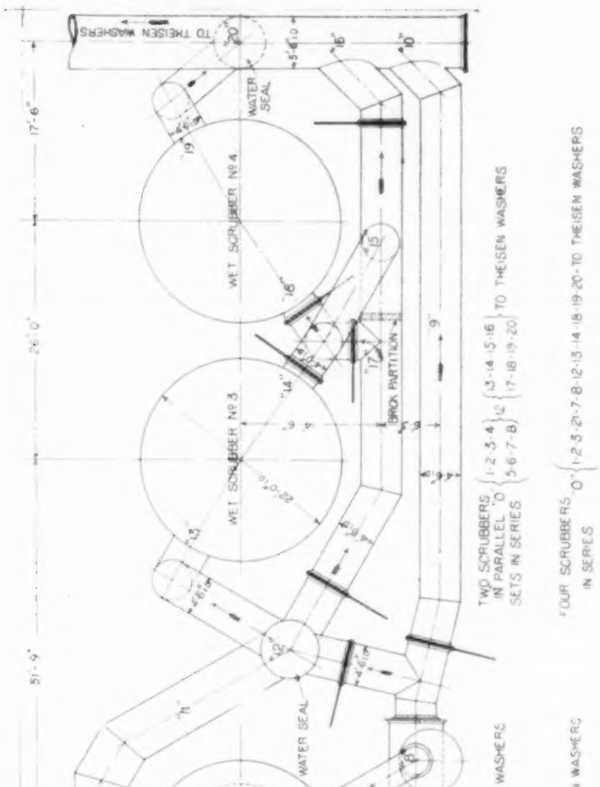


FIG. 16 FLOW OF GAS THROUGH WET SCRUBBERS

stoves, if desired. In 1909 it contained an average of not more than 0.318 grains of dust and not over 5.62 grains of moisture per cubic foot. Its purity therefore nearly complied with the requirements of European blast furnace plants where 0.5 grammes per cubic meter (0.218 grains per cu. ft.) is considered a desirable degree of cleanliness for stove and boiler gas.

IMPORTANCE OF GAS-CLEANING

47 Clean gas as delivered by the preliminary washing plant is, however, not sufficiently purified for gas engine purposes. Not so very many years ago it was thought in good faith that gas engines could operate on dirty gas, and it required years of costly experimenting to develop the art of gas purification to its present perfection, after the far-reaching importance of the problem was at last understood. Its magnitude can be better appreciated if the total quantities of gas and dust which are handled in such a cleaning plant during a certain period of time are considered. The following figures apply to the gas power plant under discussion for the year 1909.

48 The total number of kilowatt-hours generated by the gas engine installation was 50,494,100. The average heat consumption per kilowatt-hour was 17,234 B.t.u. The average heat value of the gas by calorimeter was 98.3 B.t.u. per cu. ft. The gas engines consumed therefore per kilowatt-hour 175.3 cu. ft. of gas, or in the year 1909 a total of 8,851,615,730, or nearly 9,000,000,000 cu. ft. This total quantity of gas reached the wet scrubbing plant containing on an average 1.533 grains of dust per cubic foot. There were consequently carried into the wet scrubbers during the whole year 1,938,500 lb. or 865 gross tons, of flue dust. To appreciate fully the meaning of this enormous figure it may be remembered that to haul this quantity away, a freight train of twenty gondola cars of 100,000 lb. capacity would be required. The average amount of dust in the clean gas for the year was 0.3183 gr. per cu. ft.; so that it carried 402,500 lb., or 180 gross tons of flue dust into the secondary cleaning plant. The difference of 685 tons was taken out by the wet scrubbers and carried off into the settling tanks. Expressed in per cent of the original quantity of dust, the wet scrubbers removed 80 per cent of the impurities. The Theisen gas washers further took out from the gas 176.7 tons, leaving only 3.3 tons in the fine gas, since the average amount of dust in the latter was 0.00583 grains per cu. ft. The Theisen washers had therefore an efficiency of 98 per cent, shared by clean gas main, fine gas main and gas holder.

The over-all efficiency of wet scrubbing and secondary cleaning plants was 99.5 per cent, since of the original 865 tons 861.7 tons was removed from the gas and only 3.3 tons entered the gas engines. Of the latter quantity only a small amount remained in the engine cylinders, since the bulk of the dust is swept into the atmosphere at each exhaust stroke. These figures will give a good idea of what it would mean if gas engines were operated on clean gas, not to speak of dry-cleaned gas, and yet this was attempted in the early history of the blast furnace gas engine.

SECONDARY CLEANING PLANT

49 It is generally recognized that blast furnace gas cannot be cleaned sufficiently for engine purposes without the expenditure of power, and that a satisfactory refining can only be performed in rotary gas washers on the "dynamic" principle, in contradistinction to the preliminary washing for which "static" methods are usually found to be fully adequate. Among the rotary gas washer systems on the market, the Theisen washer is considered to be mechanically well designed and very efficient. The Theisen washer installation consisted in 1909 of four washers, each of 15,000 cu. ft. per min. capacity. One additional washer has been installed recently on account of the new gas blowing-engines. Fig. 17 is an interior view of the Theisen washer building, and Fig. 18 shows the plan and elevation of this installation. The Theisen washers are arranged in two rows in a fireproof building, with the clean gas main overhead between them, and inlet pipes to the suction end of each washer. The outlet pipes pass through the building to water separators and to a ring main which delivers the gas through a 5 ft. fine gas main about 500 ft. long to the power station gas holder, and through a 4 ft. 6 in. fine gas main about 1,000 ft. long to the blowing-engine gas holder.

50 The Theisen washer, shown in sectional view in Fig. 19, consists essentially of a closed drum fitted on its outer surface with longitudinal blades arranged in spirals. This drum, supported by a shaft in two water-cooled ring-oiling bearings, rotates at high speed inside of a stationary casing of conical shape. The inlet end is equipped with suction vanes while on the discharge end an exhaust fan is firmly attached to the drum. The gas is introduced into the annular space between revolving drum and conical casing and discharged by the fan into a water separator. The operation of the washer is as follows:

51 The suction vanes draw the gas from the inlet pipe and deliver it to the longitudinal vanes, which have an inclination to the axis of

the drum so as to oppose the flow of the gas through the washer. The discharge fan at the outlet end of the drum, however overcomes this tendency and discharges the gas under positive pressure of a few inches of water. The clearance between the outer edge of the longitudinal blades and the inner surface of the stationary casing is not more than 1 in. and the gas passing through this narrow space under high pressure imparts to the water introduced at several points into the casing a movement in long spirals in an opposite direction to its own travel. This flow of water in the form of a film covering the inner surface of the stationary casing, is assisted by the conical shape of the latter, taper-

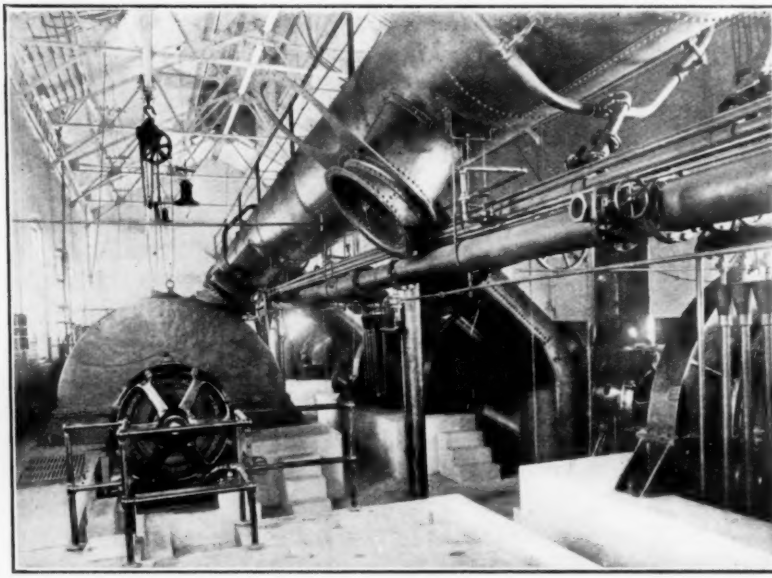


FIG. 17 INTERIOR VIEW OF THEISEN WASHER BUILDING

ing off towards the gas inlet. The surface of contact between gas and water is materially increased by wire netting, closely fitting the inside of the casing. By the intimate action of the water on the gas the dust particles are thoroughly moistened, and being weighed down by water drops, are thrown by centrifugal force into the rotating film of water, to be carried away through a seal into the sewer. The gas leaving the washer is charged with more or less moisture in the form of mist, which is removed from the gas in the Theisen washer separator, consisting principally of a removable box filled with iron shavings held in place

FOLDER No. 3

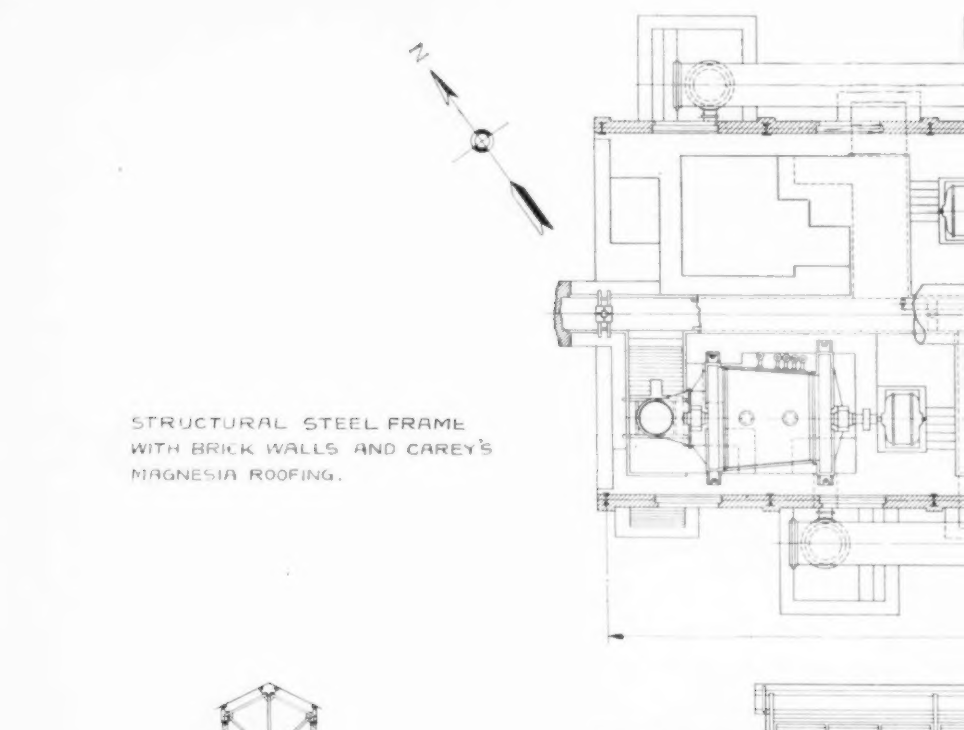
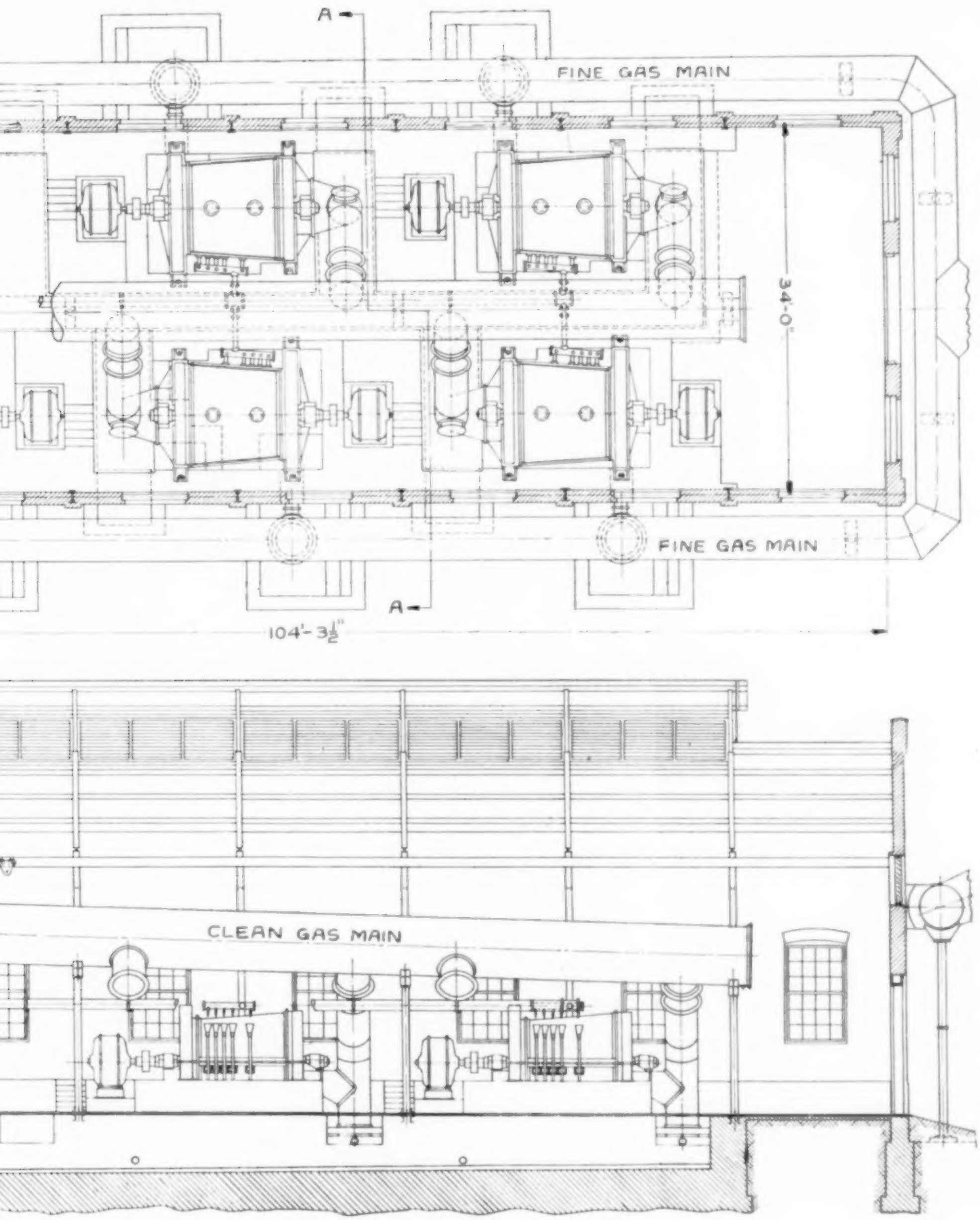


FIG. 18 PLAN AND ELEVATION OF TH



ON OF THEISEN WASHER INSTALLATION

FOLDER No. 4

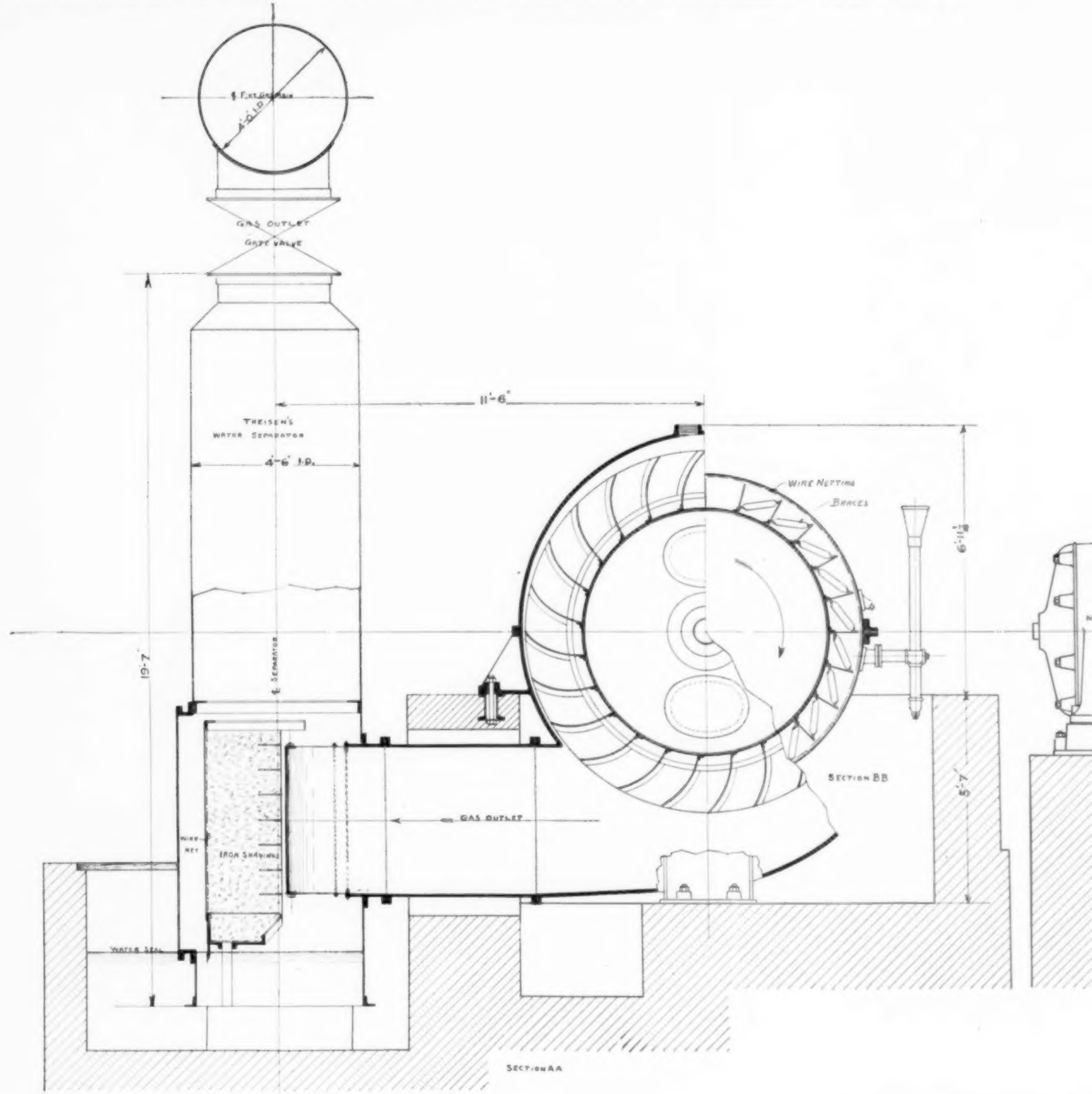
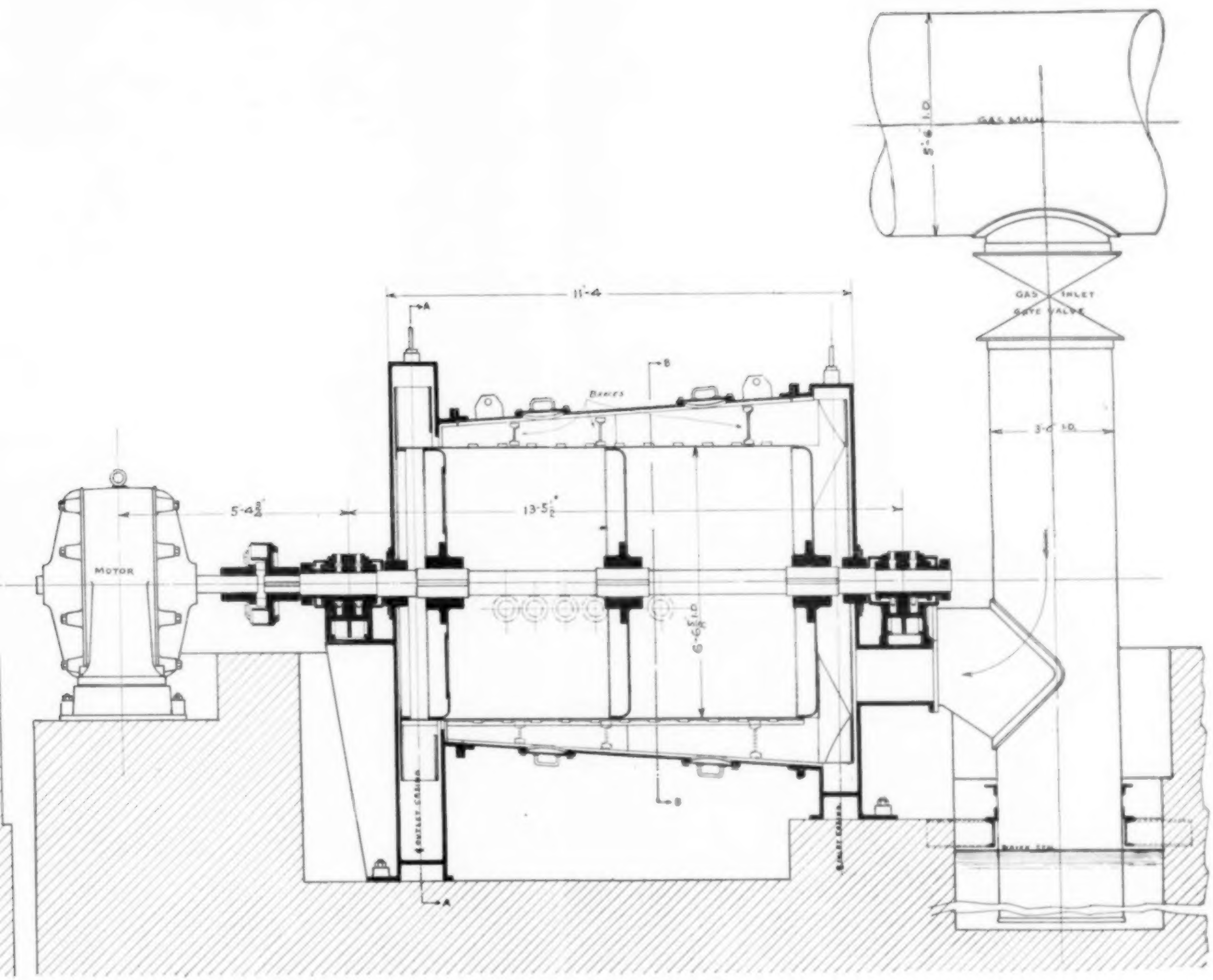


FIG. 19 SECTIONAL VIEW O



AL VIEW OF THEISEN WASHER

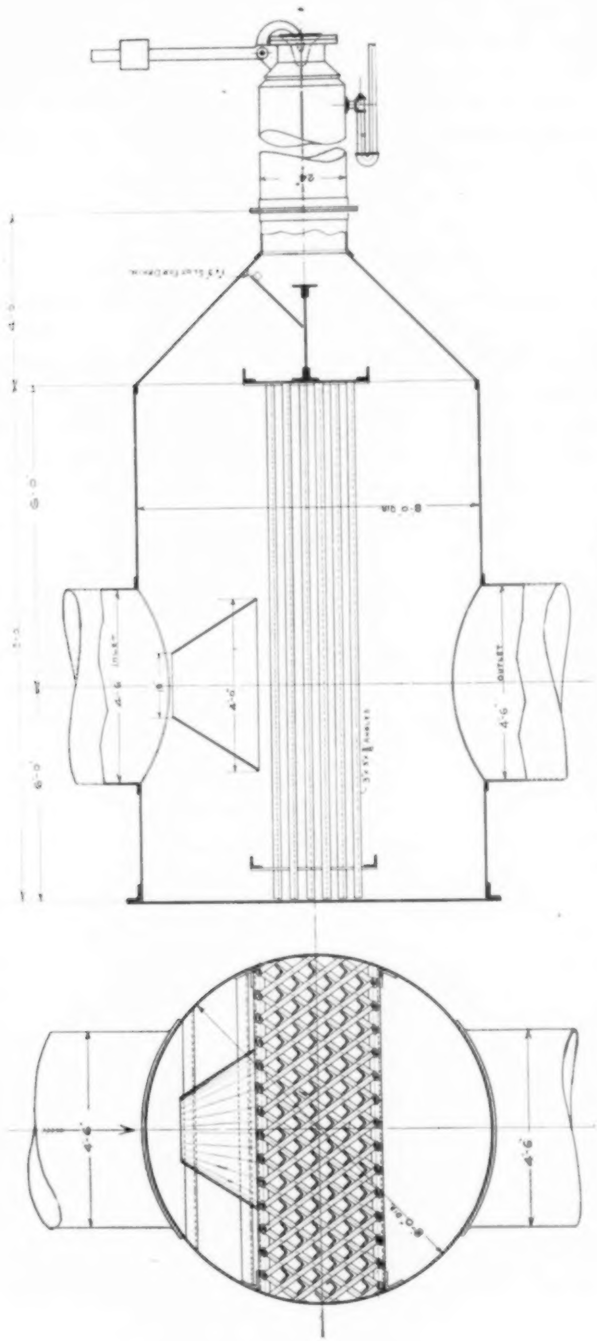


FIG. 20 WATER SEPARATOR

by wire netting. The gas striking the iron shavings with great velocity deposits its moisture, and as it has to reverse its direction it cannot pick it up again, but leaves the separator in a comparatively dry condition. Gate valves serve for regulating the quantity of gas entering the different washers, and for shutting off any washer without interfering with the operation of the plant.

52 The two gas holders, of 100,000 cu. ft. capacity each, were installed primarily to give a constant pressure of about four inches of water column at the gas engine throttle. Incidentally, however, they serve as reservoirs and as water separators. Since a gas holder was originally not contemplated for the gas blowing-engine installation an additional water separator, shown in Fig. 20, was provided in the fine gas main to the gas blowing engine house, the design of which is based on a well-known principle. A number of angle irons serve as baffles, dividing the gas into a number of streams which are forced to change their direction several times while passing through the rows of angle irons. The latter are placed "straddling" similar to the wooden slats in the wet scrubbers. A bell valve at the end of a long pipe serves to remove the accumulated water. Each gas holder is of the single-lift type, with bell 59 ft. 6 in. in diameter by 36 ft. high. Both holders have separate gas inlet and outlet pipes to obtain continuous circulation in the holder and prevent the pocketing of stale gas. While in the power station holder this idea was carried out to the extent of having inlet and outlet pipes at opposite ends of one diameter, the blowing-engine gas holder has these pipes side by side, but with the inlet turned a little to impart to the gas a rotating motion. Inlet and outlet pipes can be used as water seals to shut off the gas holder in case of necessity. To prevent the possible collapse of the gas holder bell, in case the supply of gas should be interrupted and a vacuum created underneath the holder bell, a disc valve supported by chains from the holder crown is located exactly above the mouth of the outlet pipe as shown in Fig. 21. When the holder bell descends until it rests on its landing beams, this valve will close the outlet opening, preventing a vacuum under the bell.

53 The discharge pressure of the Theisen washers is about 8 in. higher than the pressure on the suction side, and as the latter is quite variable, the former will also vary within considerable limits. It is of course possible to regulate the pressure in the fine gas main by means of the gate valves arranged in the Theisen outlets; but since these pressure variations are almost continuous, the gate valves would have to be adjusted by the operators practically all the time, unless it

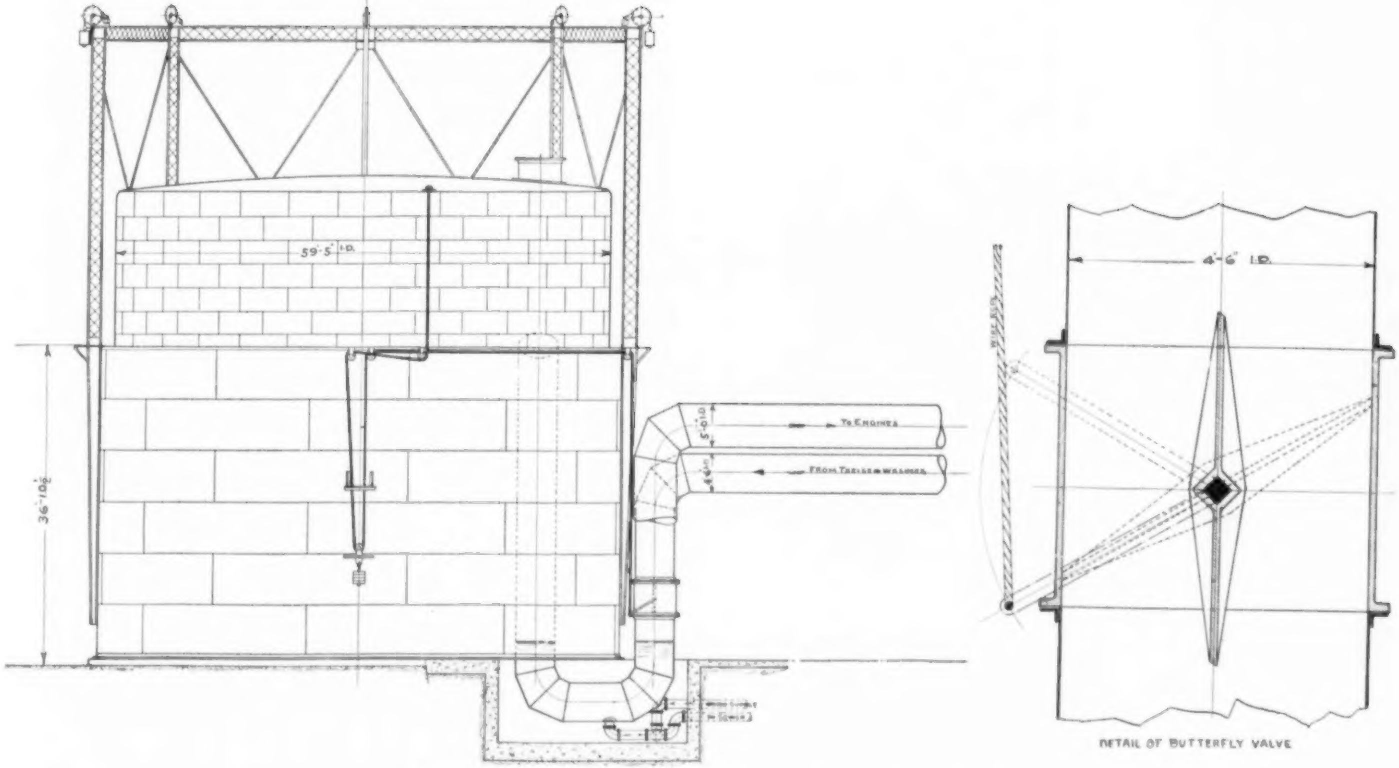


Fig. 21 Gas Holder and Automatic Butterfly Valve

were desired to resort to the complication of electrically operated valves under automatic control of the gas pressure. To simplify this necessary regulation a butterfly valve was installed in the inlet pipe to each gas holder and operated by the holder bell itself by means of a wire cable fastened to it and carried over a system of pulleys as shown in Fig. 21. This device works in the following manner:

54 When the gas holder is empty, the butterfly valve is wide open and the counterweight hanger *H* with weights *W* is in its bottom position. When the bell ascends, hanger *H* rises without affecting the butterfly valve until bumper bracket *B* is reached, which prevents the further travel of the hanger. The movable pulley *P* now becomes stationary, and the rising holder bell acts through cable *C* on the butterfly valve, throttling and finally tightly closing it, the effect being precisely the same as if the gate valves on the Theisen washer outlets had been throttled or closed. The washers continue to operate, but cease to deliver until the descending gas holder bell again opens the butterfly valve. This action is perfectly automatic and it is impossible for more gas to enter the gas holder than is being taken out, so that any number of gas engines can be started or shut down at any time without the slightest adjustment at the Theisen washers.

55 Without this automatic regulator this is what would happen: The weight of the gas container gives a constant pressure of 4 in. of water column in the outlet pipe and the Theisen washers deliver a constant quantity of gas as long as the gas pressure on the suction side is constant. With a certain number of gas engines in operation, and the gas demand equal to the gas supply, the gas holder bell will float in a certain position. If, however, one or several engines are stopped, the gas demand will decrease and as the gas supply remains constant the bell will rise into its top position, determined by the height of the water seal in the holder tank. Any further rise will break this seal, causing gas to escape from underneath the holder bell. This will continue until more engines are started and the gas demand is again equal to the gas supply, or until the Theisen outlets are sufficiently throttled to reduce the quantity of gas delivered. The disadvantages are obvious. Not only will the breaking of the seal cause large quantities of water to be thrown out, but the escaping gas, aside from being unnecessarily wasted, will dangerously foul the surrounding atmosphere. The automatic butterfly valve, balancing perfectly the gas demand and the gas supply, eliminates these troubles very effectively.

56 A by-pass line permits the operation of the gas engines directly if for any reason the holder is out of commission. Each holder de-

livers the gas into a large main located on the outside of the gas engine buildings, as shown in Fig. 22. Individual branch pipes lead the gas to each engine, which can be isolated from the gas main by bell valves or water seals, as shown in Fig. 23*a*. The gas pipes leading to the engines, as well as all gas mains, are carefully drained by automatic overflows, an important feature.

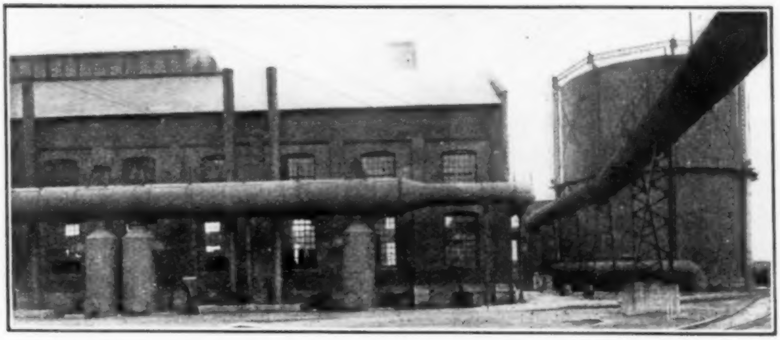


FIG. 22 EXTERIOR VIEW OF POWER HOUSE; GAS HOLDER AND GAS RECEIVERS

PERFORMANCE OF THE GAS-CLEANING PLANT

57 The physical qualities of the gas of importance from the standpoint of gas engine operation, are its pressure, temperature, dryness and cleanliness. These conditions, and particularly the last, if ascertained and suitably recorded at various stages of the cleaning process, are valuable indicators of the efficiency of the gas-cleaning plant; very few blast furnace plants, however, pay sufficient attention to their regular routine determination. As a rule tests are being made and results recorded only so long as the gas engine installation is new and therefore of all-absorbing interest. Particularly if the operation of the plant seems satisfactory, the interest is soon lost and the plant is left entirely to the care of the operating men, who soon are the only authorities on the machinery in their charge. The knowledge that can be obtained from them is of questionable value, as it is often based on good memory only, and gathered in hit-and-miss fashion. All operative results of a gas power plant should be recorded with as thorough care as is usually afforded the operation of steam plants, or even more, since the gas engine is more susceptible to variations in the quality of its fuel.

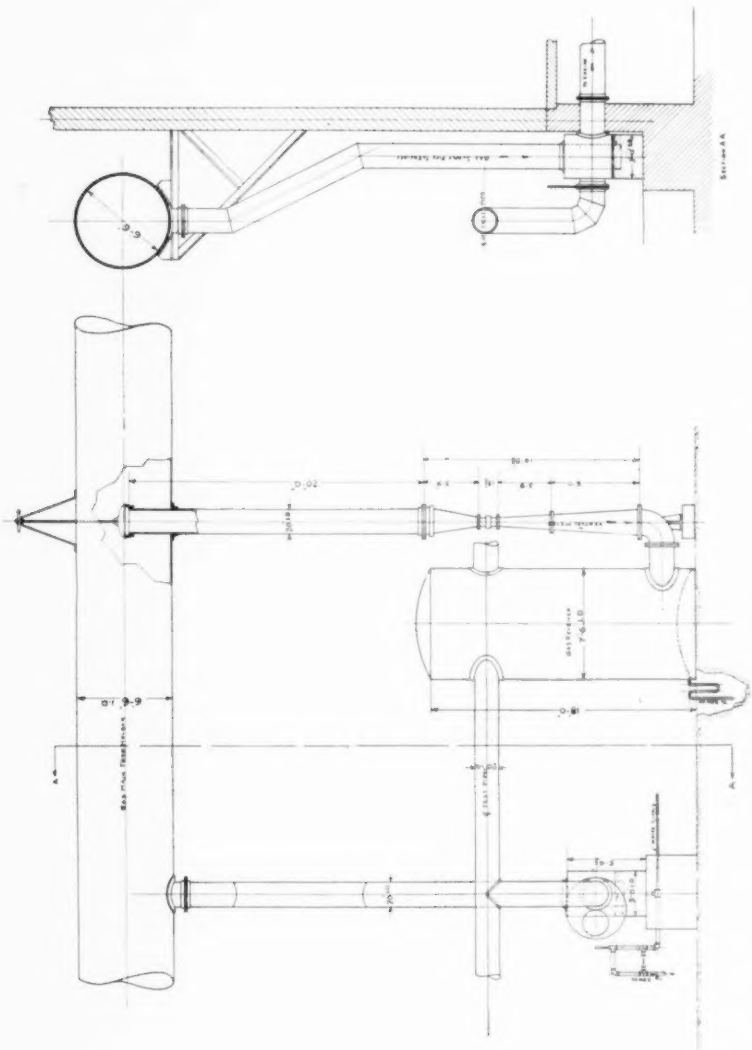


FIG. 23a GENERAL ARRANGEMENT OF TEST PIPING

58 The question is often asked, what can be the advantage of keeping exact records if the recorded results vary comparatively little during the year's operation, and whether the game is worth the candle, assuming that the expenditure is far in excess of the benefit derived. The questioner overlooks, however, that only by keeping such records can it be determined whether the conditions really are generally uniform; and that this uniformity is in many instances due to the careful watching and recording of the phenomena involved. Besides it was found, in over two years' experience at the plant under discussion, that the expense of "keeping the finger on the pulse" of the gas power plant is so small and so easily absorbed that it is insignificant. The expense connected with the maintenance of a special gas laboratory, for instance, has never as yet noticeably increased the cost of pig iron, and the three-hourly readings at the gas cleaning plant are being taken, without additional expense, by the operators themselves, who are assisted and checked in their work by recording instruments installed wherever expedient. It has been found, too, that the installation is being given much more care by the operators, since they are compelled to go over the whole plant on regular beats, in order to take the various readings. The general appearance of the gas cleaning plant shows unmistakably the influence of this continuous inspection. It is only natural that the operators should themselves become interested in their readings and compare the results from day to day. They soon make changes and improvements in the equipment, of their own accord, and will operate the plant at a much higher standard of efficiency.

59 Fig. 6 and Fig. 24 show two of the standard record forms, which are self-explanatory. The data collected on the various report sheets are tabulated and plotted on charts in the engineering department so as to show the daily, monthly and yearly averages. These records are very valuable from an operating point of view. The economy of a gas-purifying plant, for instance, is dependent on a number of elements, among which the plant efficiency is not of least importance. It involves the question of total cost per unit of gas, of electric light and power consumed in the plant, of operating labor, labor and material used in repairs, and lubricants in relation to the degree of cleanliness and the amount of moisture obtained by this total expenditure in the same gas unit. The majority of these elements can be controlled when the variations to which they are subjected are known, by voluntary or involuntary changes, but this knowledge can be acquired only through close and continuous observation.

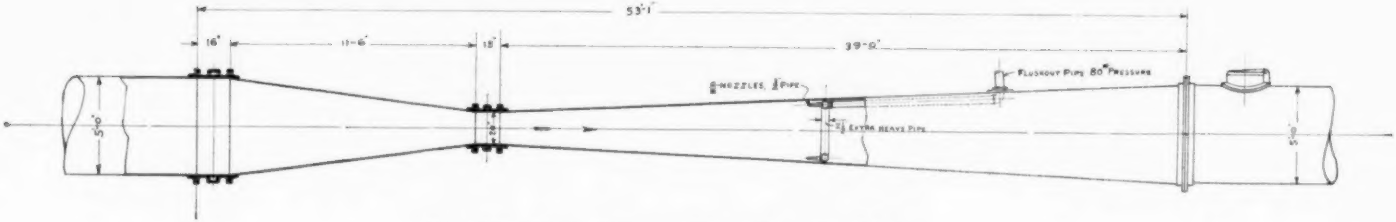


FIG. 23*b* TEST PIPING—60-IN. VENTURI METER

PRESSURES AND TEMPERATURES-DAILY REPORT

FOR 24 HOURS ENDING 6 A.M. Thursday, October 7th, 1909.

GAS TEMPERATURES (°F) PRESSURES ("OF WATER)

TIME		Entering Gas Cleaning Plant T	Entering Gas Cleaning Plant P	After Wet Scrubbers 1 T	After Wet Scrubbers 2 T	After Wet Scrubbers 3 T	After Wet Scrubbers 4 T	Before Theisen Washers T	Before Theisen Washers P	After Theisen Washers T	After Theisen Washers P	A. C. Sta. Gas Holder In T	A. C. Sta. Gas Holder Out T	B. Engine Gas Holder In T	B. Engine Gas Holder Out T
DAY TURN	8 A.M.	218	17.0	—	—	77	64	68	16	68	22	67	65	—	—
	11	196	16.5	—	—	83	74	76	16	76	21	75	74	—	—
	2 P.M.	204	14.0	—	—	82	75	77	13	77	19	79	77	—	—
	5	208	15.5	—	—	82	72	74	15	74	21	74	72	—	—
NIGHT TURN	8	184	16.0	—	—	76	68	72	15	73	18	73	68	—	—
	11	202	14.0	—	—	78	70	73	13	74	16	72	68	—	—
	2 A.M.	200	13.0	—	—	76	67	71	12	71	15	68	65	—	—
	5	180	14.0	—	—	72	66	70	13	68	16	67	64	—	—
AVERAGES		199	15.0	—	—	78	69	73	14	73	18	72	69	—	—

WATER TEMPERATURE; READING OF AMMETER ON THEISEN WASHER MOTORS (AMPERES PER TERMINAL)

TIME		Pump Suction	Wet Scrubber Waste 1	Wet Scrubber Waste 2	Wet Scrubber Waste 3	Wet Scrubber Waste 4	Waste Theisen Washers	Air Temperature	Ammeter 1	Ammeter 2	Ammeter 3	Ammeter 4	Ammeter 5	Ammeter 6	Ammeter 7	Ammeter 8	Water Consumption Gal./Min. Scrub.	Water Consumption Gal./Min. Th'sn	No. of Electric Engines in Operation	No. of Blowing Engines in Operation
DAY TURN	8 A.M.	67	—	—	96	69	70	66	—	—	155	155	—	—	—	—	1400	300	3	—
	11	69	—	—	98	71	72	76	—	—	155	155	—	—	—	—	1400	300	3	—
	2 P.M.	74	—	—	104	76	77	72	—	—	155	155	—	—	—	—	1400	300	3	—
	5	74	—	—	104	76	77	70	—	—	155	155	—	—	—	—	1400	300	4	—
NIGHT TURN	8	70	—	—	100	72	73	69	—	—	155	155	—	—	—	—	1400	300	4	—
	11	72	—	—	102	73	74	68	150	—	—	155	—	—	—	—	1400	300	4	—
	2 A.M.	70	—	—	98	71	72	67	150	—	—	156	—	—	—	—	1400	300	4	—
	5	70	—	—	98	71	72	65	150	—	—	155	—	—	—	—	1400	300	4	—
AVERAGES		71	—	—	100	72	73	69	150	—	155	155	—	—	—	—	1400	300	—	—

SUPPLIES RECEIVED

OBSERVATIONS

Started Bibus pump at 8:00 A.M. shutting down centrifugal pump.
Repacked centrifugal pump and started again, shutting down Bibus pump at 11:00 A.M.
Started Theisen washer No.1 at 3.00 P.M. shutting down washer No.3 on account of bricklayers working around No.3 motor.

Martin Mullin — Day Operator
Mike Smolinski — Night Operator

FIG. 24 DAILY REPORT FORM FOR GAS-CLEANING PLANT

60 Suppose, for example, the dust determinations of a certain day, or of several consecutive days, show a much higher amount of impurities in the engine gas than usual. Steps to remedy this condition may be taken immediately. As a rule an increased amount of water in the preliminary washing plant, or on the Theisen washers, will have the desired effect; or an additional washer can be started, thereby decreasing the load on each unit and giving the gas an additional scrubbing. Without the daily record this increase in dust might not be noticed until trouble arose in the engines; or a clogging of the wet scrubbers or of the gas flues might not be noticed until the effect of a restricted gas passage was shown in the reduced output of the power plant.

61 For example, on consulting the daily records in September and October 1908, it was noticed that the gas pressure between the two wet scrubbers was considerably lower than that in the collecting flue, a state of affairs particularly annoying at that time as another period of insufficient gas supply was on hand. It was first thought that the hurdles in wet scrubber No. 1 were clogged by dust bridging over between slats. Simultaneous readings of the pressure gages on either side showed a difference of $1\frac{1}{4}$ in. to $2\frac{1}{8}$ in. After flushing the scrubber for 30 minutes by opening wide all the top sprinklers and side flush-outs, which are situated half way between top and bottom of the scrubber, and using 1,800 gal. of water per minute, this difference did not disappear, indicating beyond a doubt that no obstruction had occurred in the scrubber. It was finally found that the scrubber inlet pipe was nearly filled with mud at the point where it turns at a slight angle into the wet scrubber shell and a heavy stream of water soon removed the obstruction.

62 In March 1909 the amount of flue dust in the dry cleaned gas increased rather suddenly from 0.56 grains per cu. ft. on March 3 to 1.53 grains per cu. ft. on March 5. The amount of water on wet scrubber No. 2 was increased from 400 to 500 gal., and decreased from 400 to 350 gal. per min. on the Theisen washers. The effect was a material improvement in the wet scrubber efficiency, while the amount of dust in the fine gas was hardly affected. This is illustrated in Fig. 25, showing the daily averages of the dust contents in the gas, etc. In this manner the total amount of water for wet scrubbers and Theisen washers as well as the relative quantities for scrubbers No. 1 and No. 2, were changed frequently during the year to conform with the demands indicated by variations in the recorded results. The methods and instruments used to obtain these records are given in Appendix No. 3.

RECORDS AND RESULTS OF OPERATION OF THE DRY-CLEANING PLANT

63 Before the existence of the dry cleaning system at the blast furnaces the two dry dust catchers in the preliminary gas-cleaning plant proved very satisfactory in operation and efficiency, and the effect of unlined gas flues and dust catchers on the reduction in temperature of the gas was greater than had been anticipated. The temperature at which the gas leaves the furnace top averages about 400 deg. fahr. with the furnaces in normal operating condition. This temperature may, however, reach 700 and 800 deg. when abnormal conditions of operation are caused by high coke consumption, irregular working, etc., and furthermore when special grades of iron, such as ferrosilicon or spiegeleisen are produced. When formerly all furnaces discharged their gas directly into the brick-lined overhead flue the temperature of the gas at the entrance of the gas-cleaning plant was considerably higher. Thus in 1908 the average temperature at the main water seal, according to Bristol pyrometer records, was as follows in degrees fahrenheit:

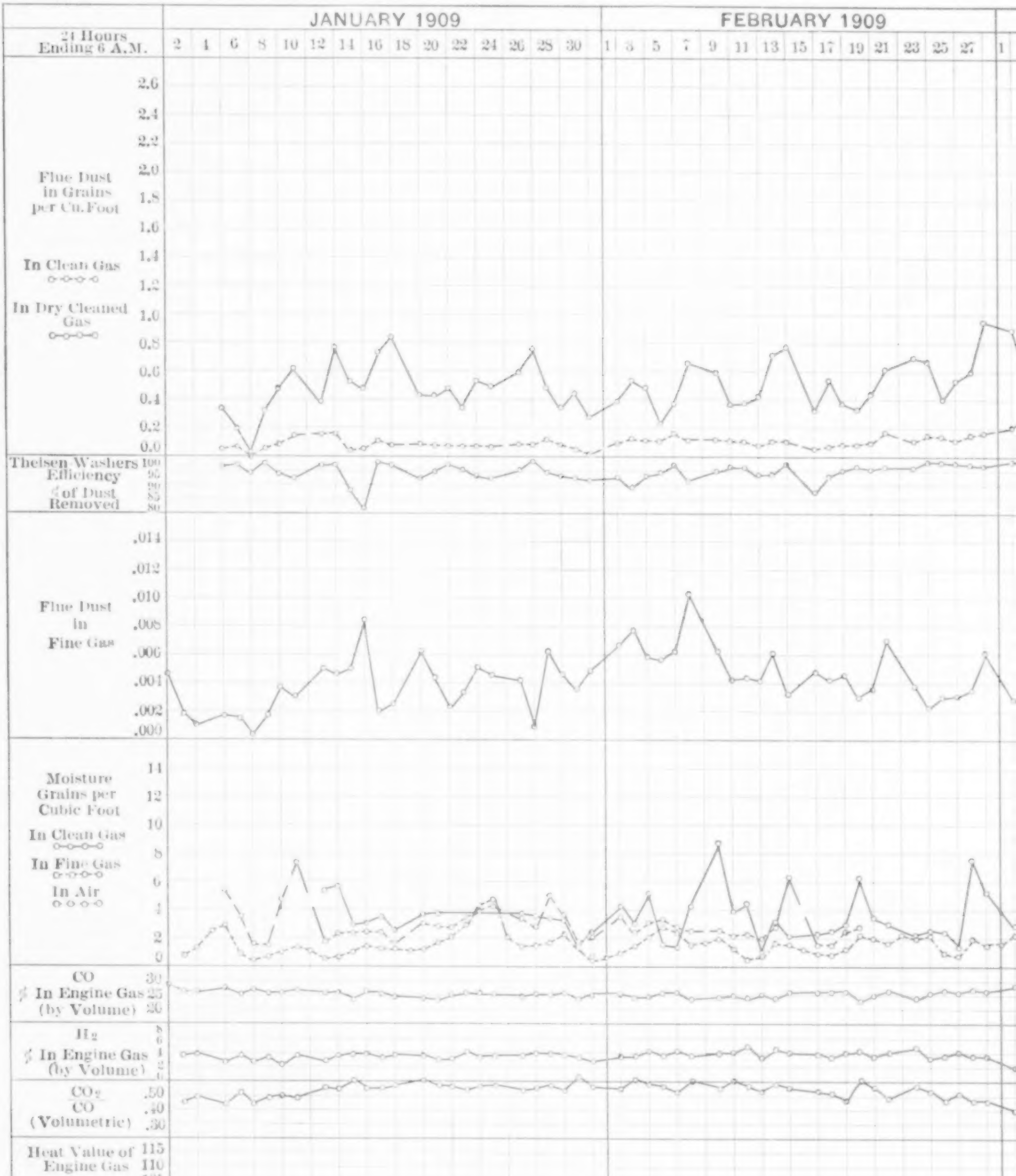
March	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg.
650	500	483	531	426	410	303	312	299	329	425

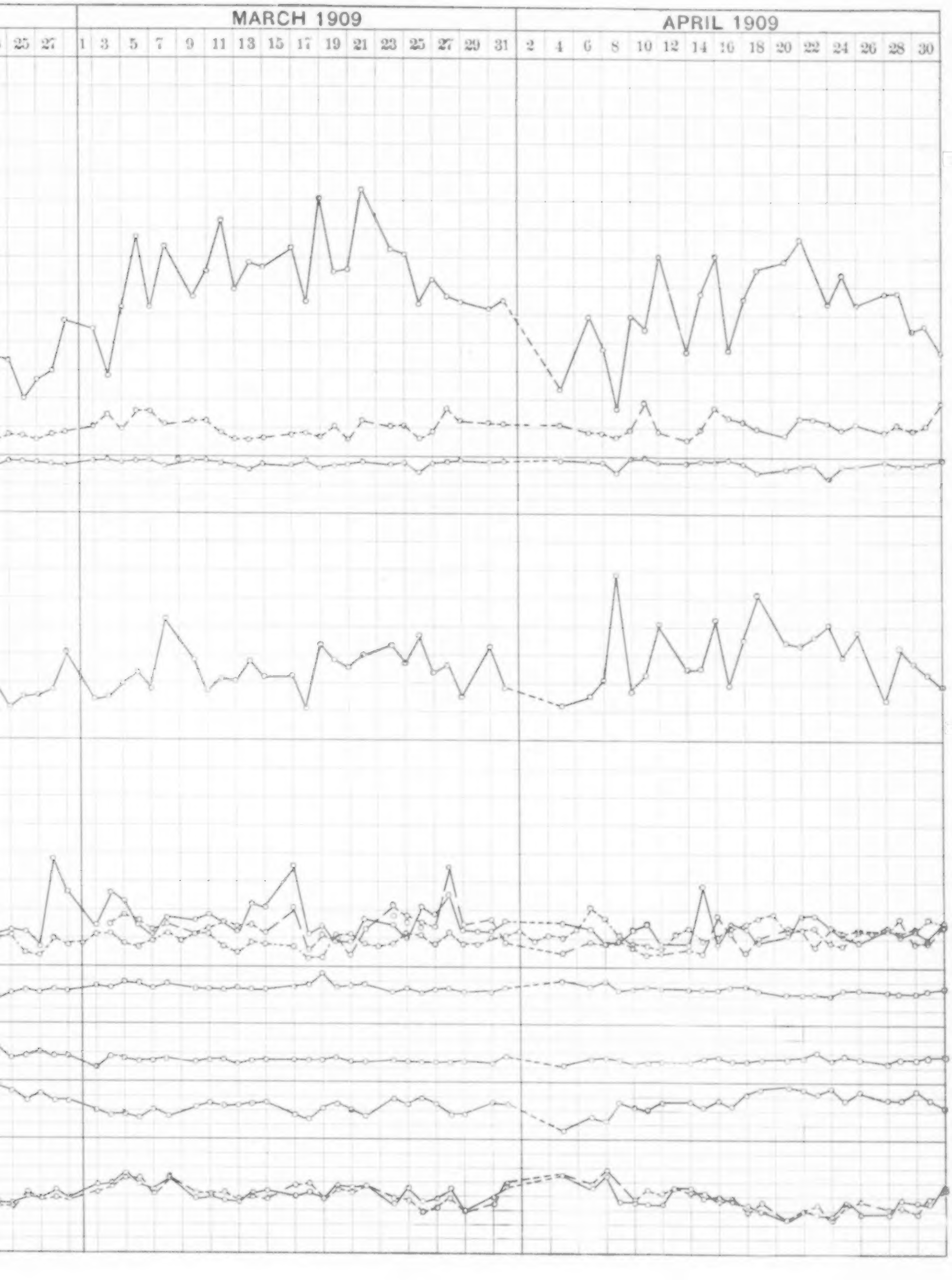
The drop in temperature of the gas, due to radiation of heat through the walls of unlined piping and dry dust catchers, was quite pronounced. See Appendix 4, Table 1. In round numbers about 50 per cent of the sensible heat carried by the gas into the dry-cleaning plant was removed by radiation.

64 An attempt was made to determine the number of B.t.u. radiated per hour per square foot of radiating surface of the dry dust catchers to obtain a basis for future calculations. For five different days the B.t.u. loss per square foot of radiating surface per degree difference in temperature per hour was 1.29; 0.99; 1.105; 1.11; 1.33; with an average for all observations of 1.165.

65 After the dry dust catcher system at the blast furnaces was put in operation conditions changed considerably, as the gas passing through the voluminous unlined dry dust catchers and the overhead gas main, the brick lining of which had been removed in April 1909, lost so much heat by radiation that it entered the gas-cleaning plant at a much lower temperature than before. This temperature is very uniform at present, averaging about 300 deg. fahr. The heat-radiating effect of the dry-cleaning plant, however, is maintained, reducing the average temperature of the gas before it enters the wet-cleaning

FOLDER No. 5


JANUARY 1909
FEBRUARY 1909
24 Hours Ending 6 A.M.
Flue Dust in Grains per Cu. Foot
In Clean Gas
In Dry Cleaned Gas
Theisen Washers Efficiency % of Dust Removed
Flue Dust in Fine Gas
Moisture Grains per Cubic Foot
In Clean Gas
In Fine Gas
In Air
CO % In Engine Gas (by Volume)
H_2 % In Engine Gas (by Volume)
CO_2 / CO (Volumetric)
Heat Value of Engine Gas B.T.U. per Cu. Ft.
by Calorimeter
By Analysis


25 27
MARCH 1909
1 3 5 7 9 11 13 15 17 19 21 23 25 27 29 31
APRIL 1909
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30

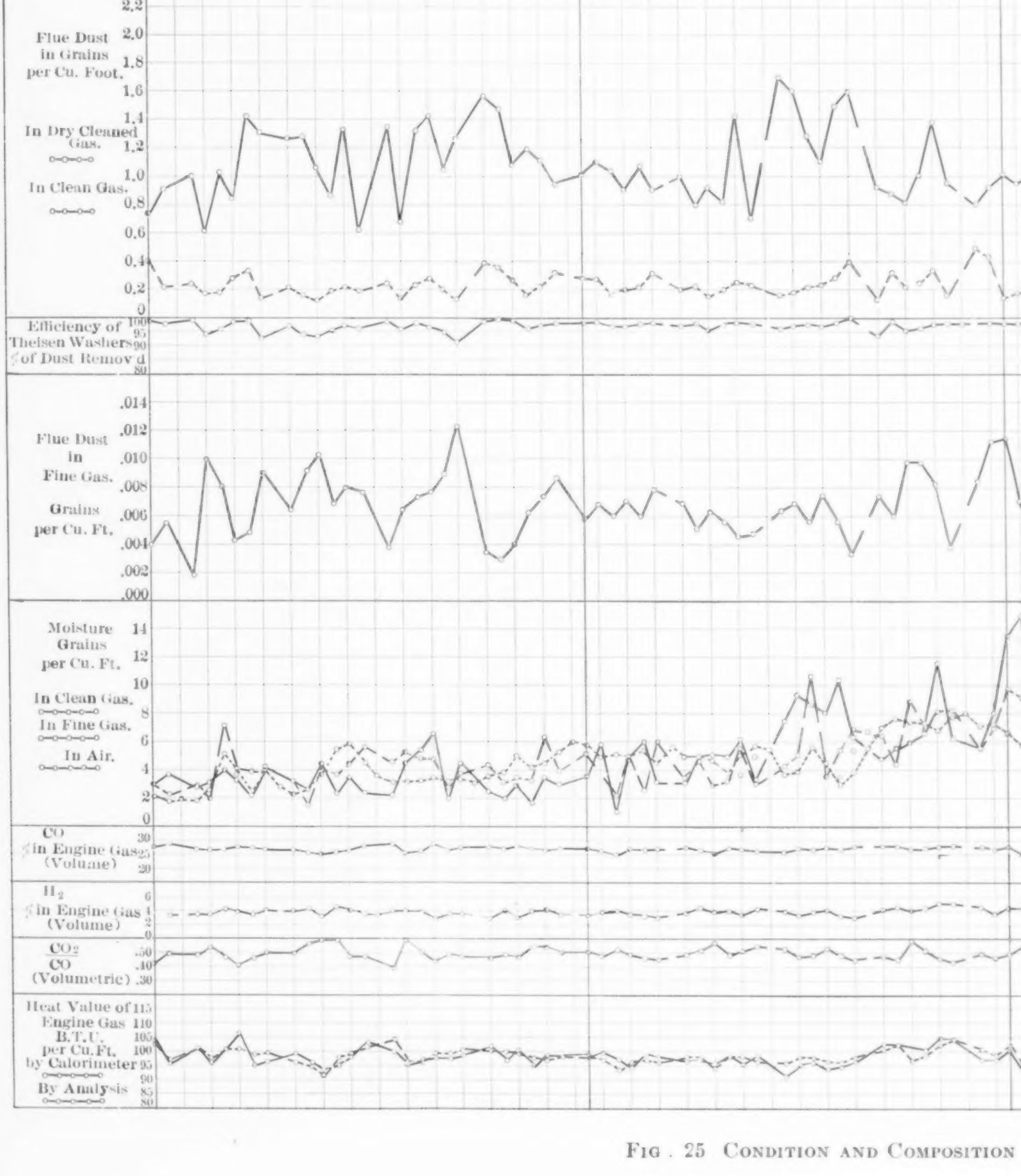
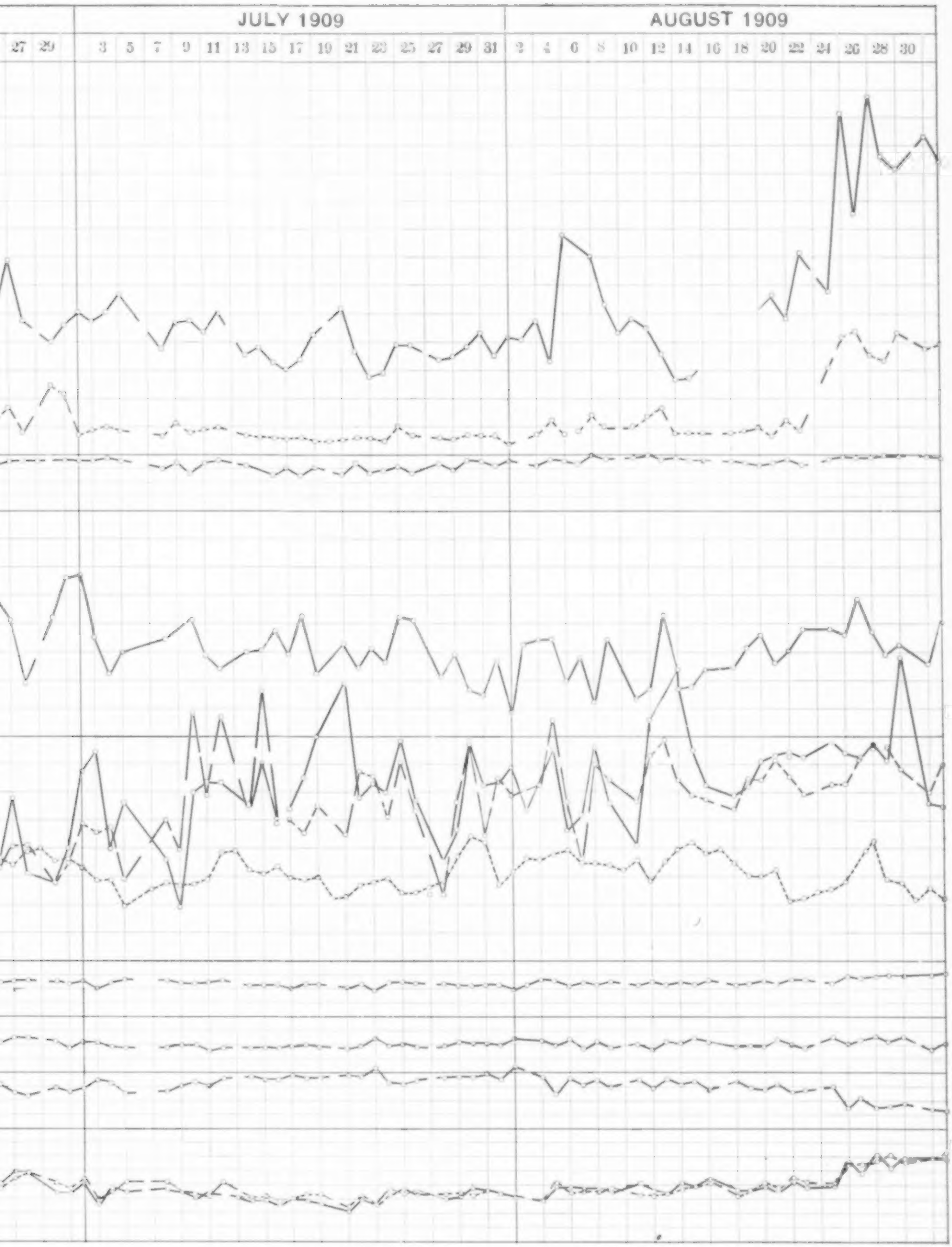
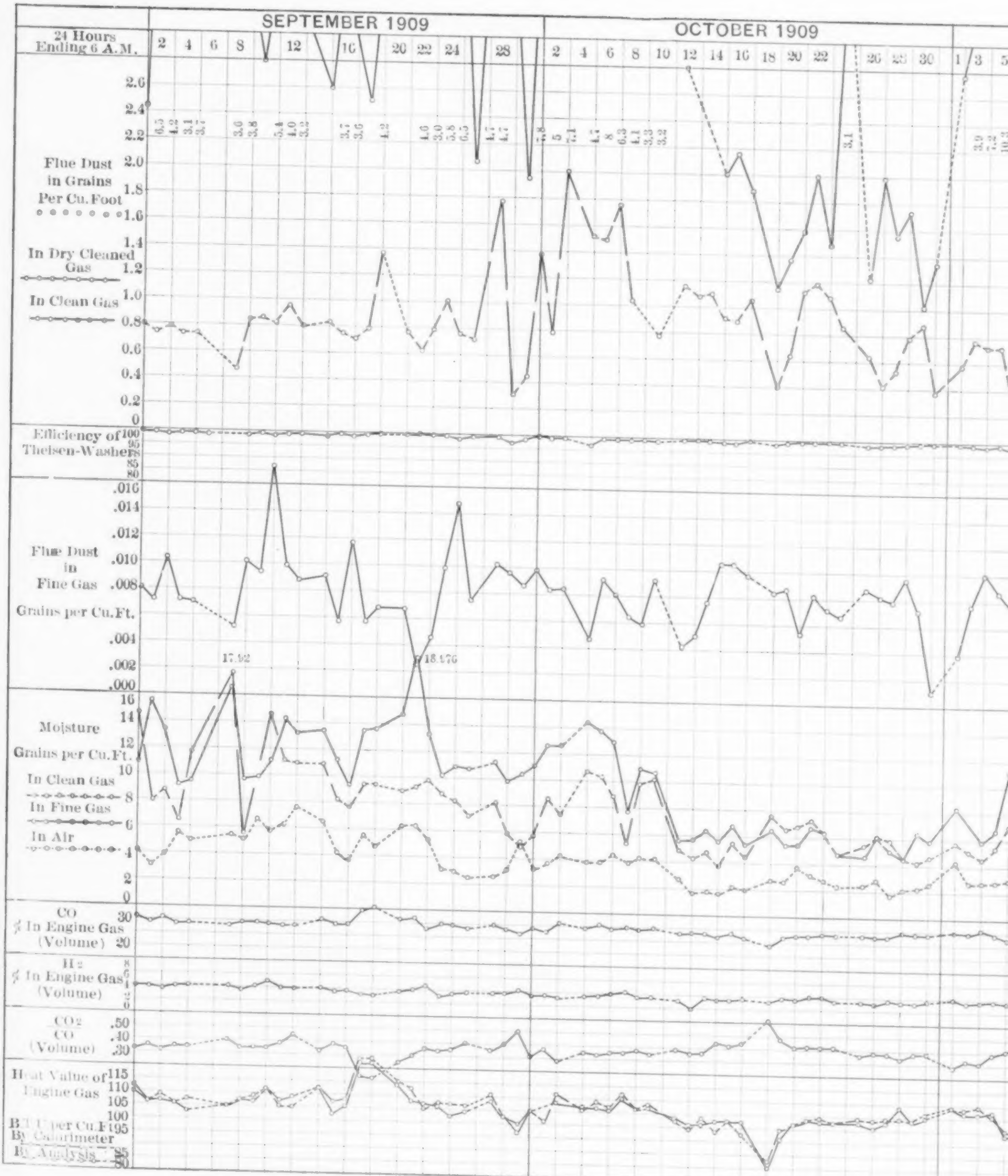


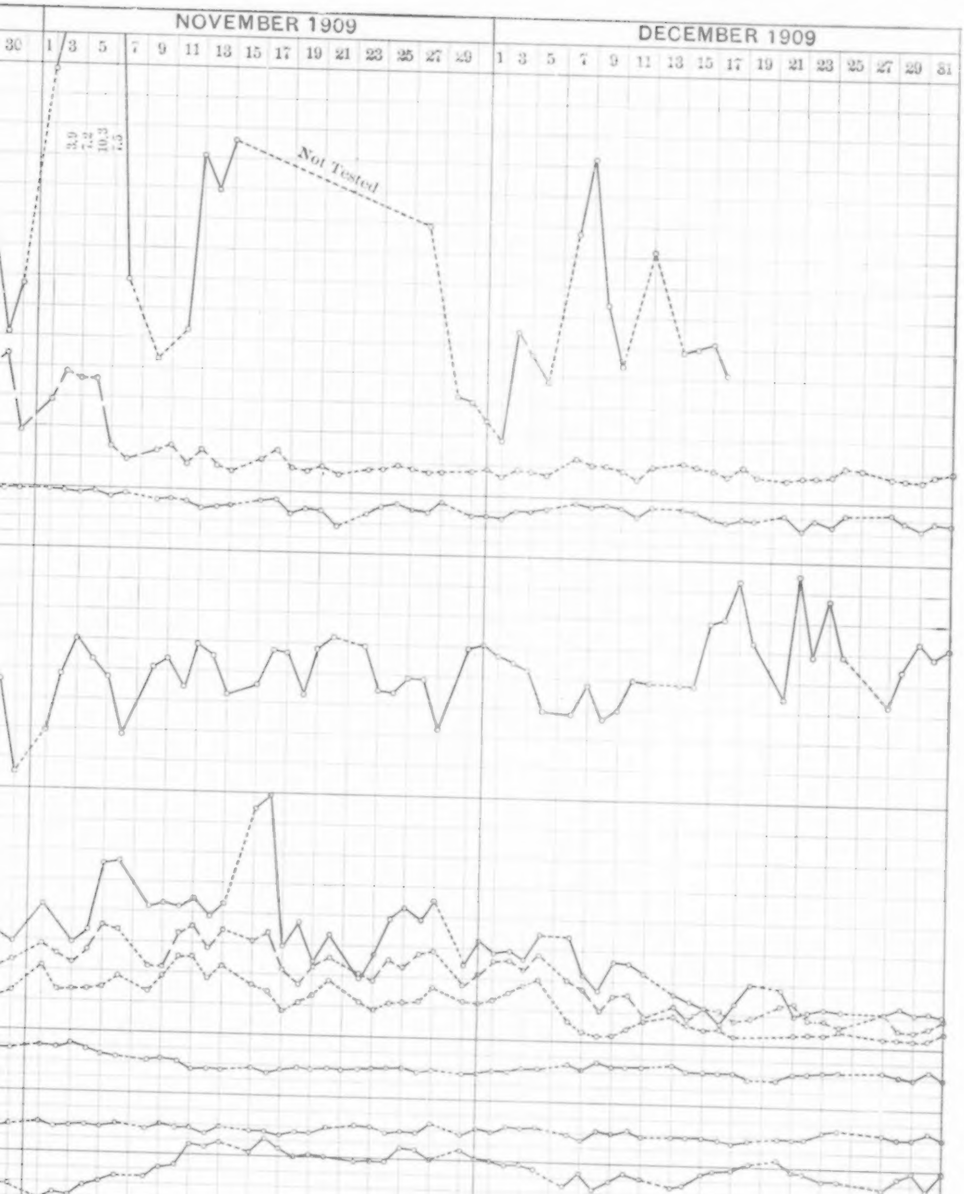
FIG. 25 CONDITION AND COMPOSITION



MPOSITION OF GAS DAILY (AVERAGES)



SEPTEMBER 1909
OCTOBER 1909
24 Hours Ending 6 A.M.
Flue Dust in Grains Per Cu. Foot
In Dry Cleaned Gas
In Clean Gas
Efficiency of Theisen-Washers
Flue Dust in Fine Gas Grains per Cu.Ft.
Moisture Grains per Cu.Ft.
In Clean Gas
In Fine Gas
In Air
CO % In Engine Gas (Volume)
H_2 % In Engine Gas (Volume)
CO_2 / CO (Volume)
Heat Value of Engine Gas
B.T.U. per Cu.Ft.
By Calorimeter
By Analysis


NOVEMBER 1909
DECEMBER 1909
30 1 3 5 7 9 11 13 15 17 19 21 23 25 27 29
1 3 5 7 9 11 13 15 17 19 21 23 25 27 29 31
Not Tested

plant about 56 per cent as shown in Fig. 26, which gives the average monthly figures for 1909. Since the cooling effect takes place without the use of water, it is obtained entirely without cost.

66 This cooling of the gas to a temperature considerably below 212 deg. in 1909 caused heavy condensation of moisture in the pipes and

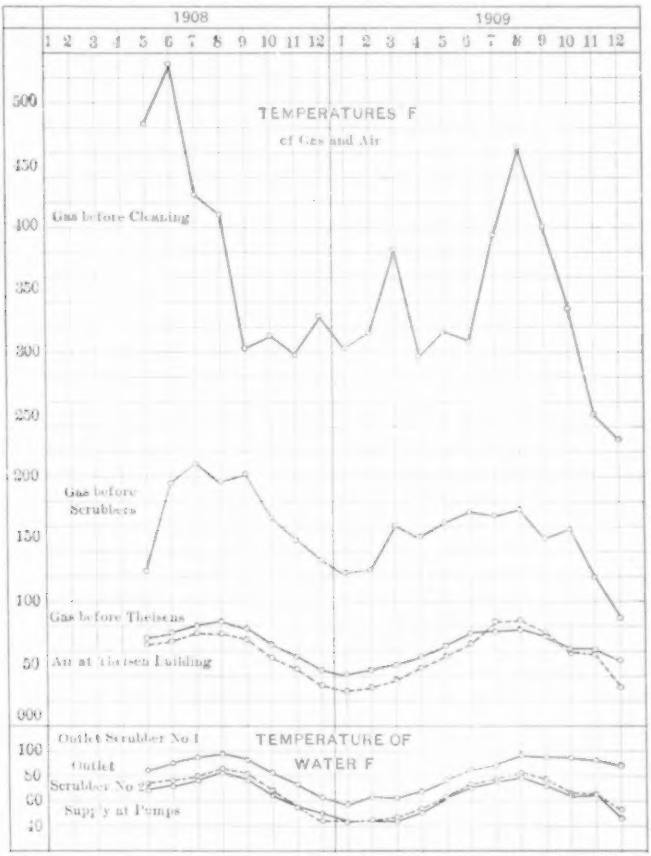


FIG. 26 TEMPERATURES OF GAS, WATER AND AIR (MONTHLY AVERAGES)

dry dust catchers. This proved to be of considerable value in the process of gas cleaning, since the finer particles of dust were weighed down by microscopical globules of water and thus more easily separated from the gas. The dust removed in the dry cleaning plant appeared mostly in the form of mud, which accumulated at all points of

change in direction of the gas flow, and especially in the dry dust catchers where the velocity was very small. The difficulties encountered in trying to remove this mud during the operation of the plant formed one of the reasons for abandoning the dry dust catchers at the gas-cleaning plant. As outlined in Appendix No. 3, several attempts to obtain reliable data concerning the dust-removing efficiency of the dry-cleaning plant proved futile, but an idea can be gathered from the fact that on an average a carload of dry dust weighing about twenty-five tons was removed every other week, while a large quantity was blown away by the wind.

67 The condition in which the gas was delivered to the next stage of cleaning is shown in charts (Figs. 25, 27 and 28) and in Appendix 4, Table 2, wherein the monthly averages as well as the daily variations of the dust contents in the dry cleaned gas are given. These curves and particularly Fig. 28, which gives the daily amount of flue dust in dry cleaned and clean gas for the period from August to December 1909, drawn to a larger scale, indicate quite violent fluctuations which are due to different operating conditions of the blast furnaces.

68 Heavy slipping will naturally increase the dust contents in the raw gas beyond measure. It has an effect very similar to an explosion, as the sudden upheaval of the stock in the furnaces causes a momentary rise in the gas pressure, illustrated on pressure chart Fig. 1. The velocity of travel of the gas through the pipe lines after a slip can easily be observed in the rapidity with which clouds of flue dust belch forth from boiler stacks, nearly 1,000 ft. from the source of disturbance, only a very few seconds after the slipping. Besides the momentary increase in the quantity of dust caused by slipping the furnaces, considerable amounts are added from the deposits of flue dust in pipe lines, dust legs and dry dust catchers, accumulated for hours and days, and disturbed by the sudden high velocity of the gas.

69 The nature of the furnace product has a great deal of influence on the quantity of flue dust produced. While Bessemer and basic furnaces produce about equal amounts in the plant under discussion, ferrosilicon and spiegel furnaces make very much more, which moreover is very fine and cannot easily be removed from the gas—especially not by dry cleaning alone. Thus for instance a sudden increase of 270 per cent in the dust contents in the dry cleaned gas occurred in March 1909, due to the following cause: For several months previous the raw material charged into the furnaces had been considerably "watered" to reduce top temperatures and flue dust losses. On

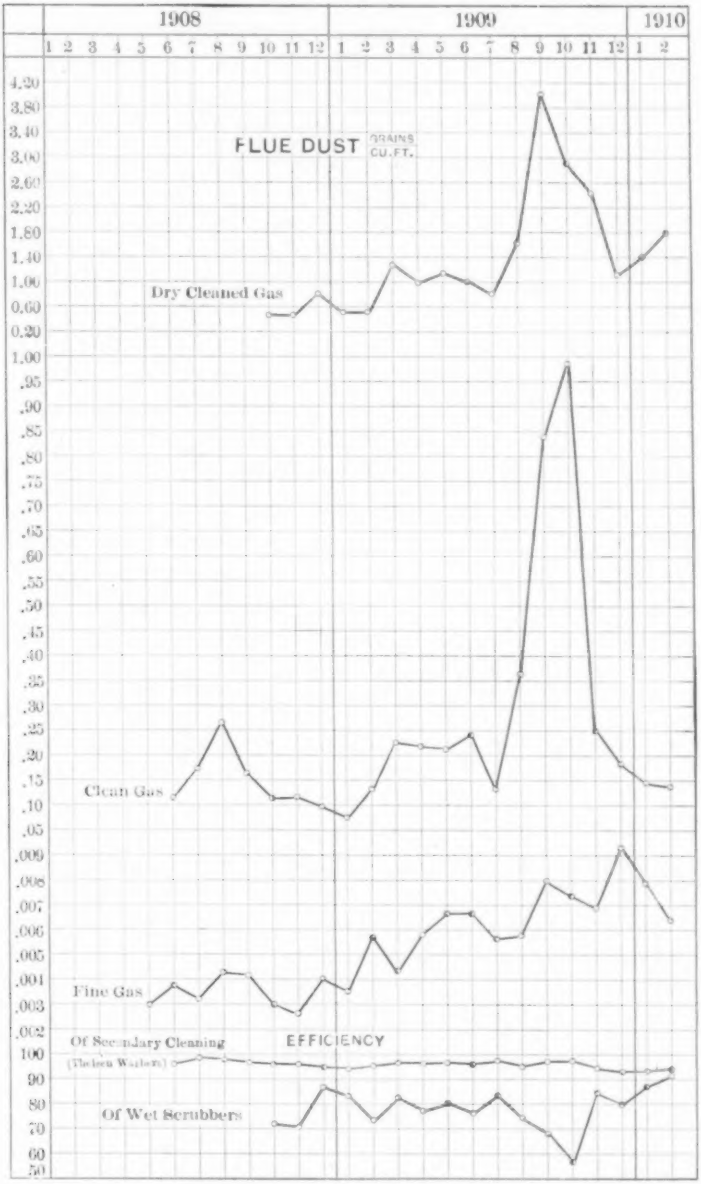


FIG. 27 FLUE DUST IN GAS AND EFFICIENCIES OF CLEANING PLANT (MONTHLY AVERAGES)

February 28, 1909, the watering of the stock was suddenly discontinued. The result was an increase of nearly 100 per cent in the dust contents in the dry cleaned gas, as shown in Fig. 25. It begins with the day ending March 4 and shows great fluctuation, while after March 24 a sudden decrease occurs and the amount of dust for the rest of the month shows considerably more uniformity.

70 The reduction was due to the fact that on March 24 the watering of the stock was resumed in a moderate way. The average amount of flue dust in the dry cleaned gas in February was 0.4787 grains, against 1.2951 grains per cu. ft. in March, the corresponding increase of dust in clean gas being from 0.1224 grains in February to 0.2238 grains per cu. ft. in March, or about 200 per cent. The quantities of gas cleaned per minute were almost exactly the same, namely 14,765 cu. ft. in February and 14,717 cu. ft. in March, so that the records are directly comparable. It cannot be estimated even, how much dust the raw gas contained during March, but when the dry dust catchers were opened for examination on April 3 it was found that the mud in the bottom cones had accumulated to the umbrella, while large dust and mud deposits were discovered in the piping even, the latter having lost its self-cleaning qualities due to the nature of the deposits, which in their muddy condition refused to slide down into the dust legs.

PERFORMANCE OF WET-SCRUBBING PLANT—COOLING AND CONDENSING EFFECT

71 The preliminary wet-scrubbing plant was particularly successful and efficient, and the operation of the wet scrubbers has been continuous ever since starting in November 1907. Several examinations of the wet scrubbers took place, at times when the gas power station was shut down entirely, and it was invariably found that both were in perfect condition. Aside from a thin coating of slimy flue dust, which seems to have penetrated into the pores of the wood, there was no sign of any deposit on or around the hurdles. Since oxygen is practically entirely absent, rotting of the woodwork is impossible. From observations of the condition of the wet scrubbers the conclusions may be drawn that it is unnecessary to be particular in selecting the quality of lumber for manufacturing the slats and that dressing it all over could possibly be dispensed with. The distance between the slats could be made very much smaller than in the two original wet scrubbers, and a spacing of about three inches was adopted for

FOLDER No. 6

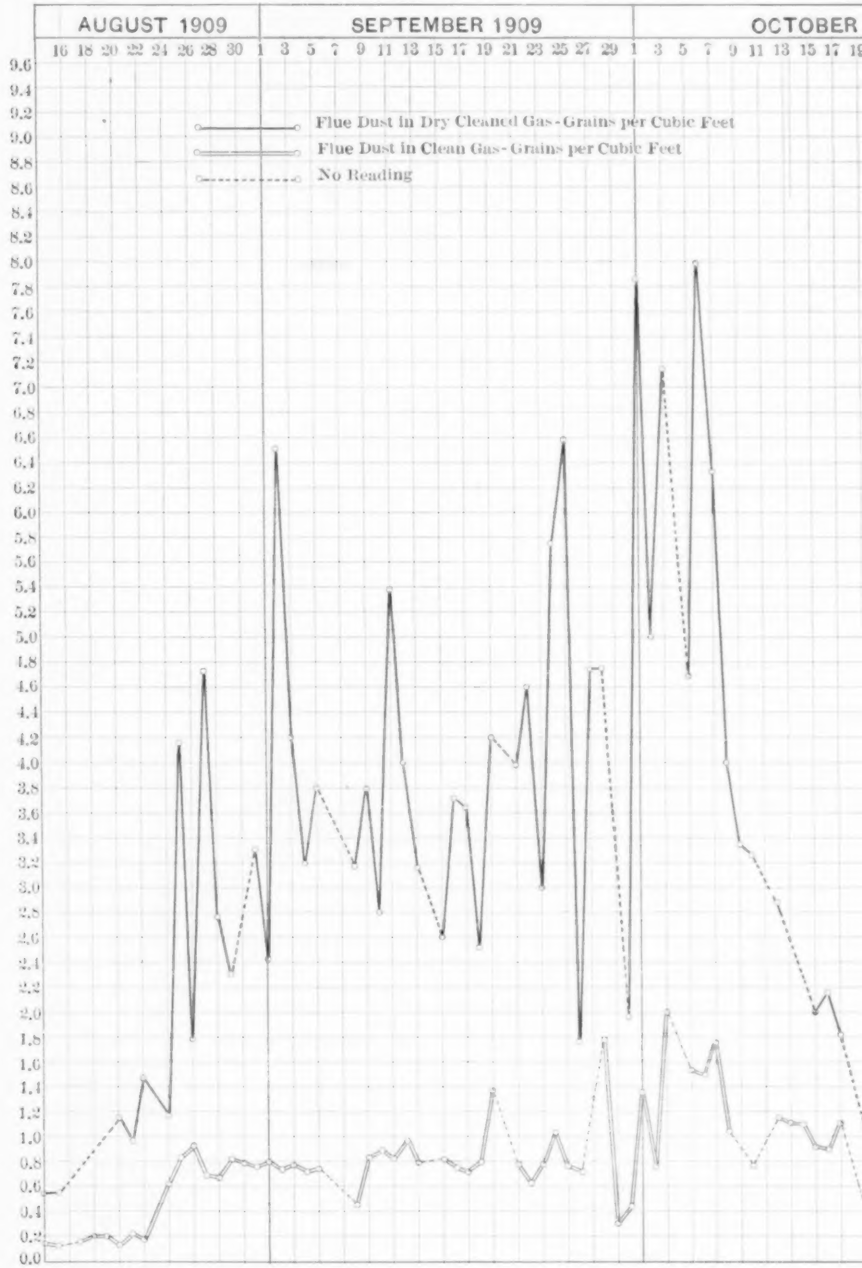
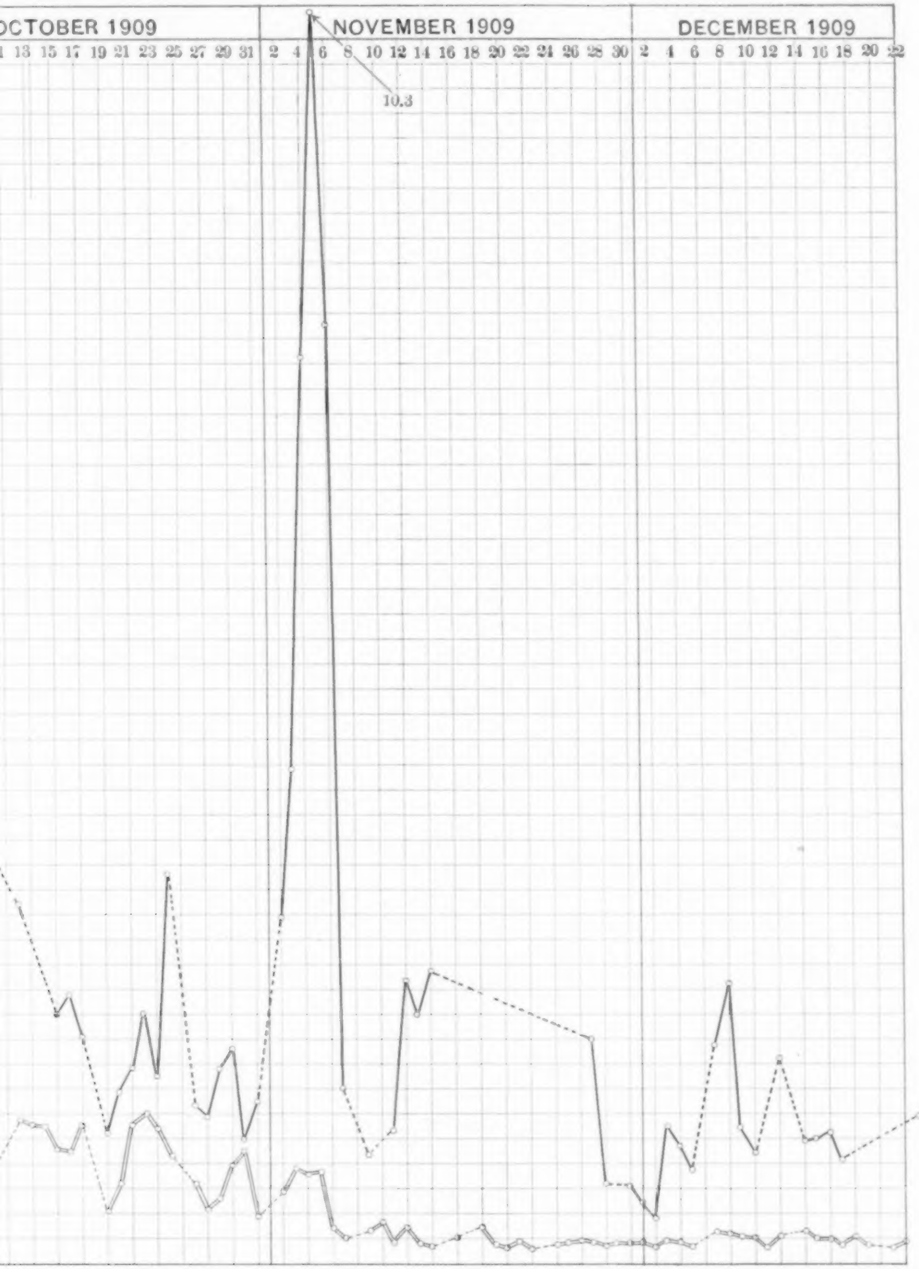


FIG. 28 FLUE DUST IN DRY



T IN DRY CLEANED AND CLEAN GAS

the two new ones. No difficulties of any description were encountered in the operation of these scrubbers, and the overflow arrangement from the water sealed basins, shown in Fig. 11*b*, as well as the disposal of the dirty washing water never caused any trouble.

72 In Fig. 26 and in Appendix 4 (Table 3) are shown the effects of wet cleaning upon the temperature of the gas and the corresponding fresh and waste water temperatures. A comparison of the curves shows that the gas was cooled practically to water temperature, and records show that this cooling action is nearly limited to the first wet scrubber, as the temperature of the gas between the wet scrubbers was only a few degrees higher than the clean gas temperature. The average temperature of the clean gas for the first half year was 54.7 deg., while the temperature of water supply was 53.3 deg. for the same period. For the second half of 1909 these temperatures were 67.4 and 66.7 deg. respectively. The yearly average temperature of the clean gas was 61.1 deg., while the yearly average temperature of the water supply was 60.0 deg. The temperature of the waste water from scrubber No. 1 was on an average 20 deg. higher than the temperature of the fresh water, while the water from wet scrubber No. 2 did not exceed the average water supply temperature more than 1.7 deg. fahr. The first wet scrubber naturally removed the bulk of the dust, as was indicated by the muddy, black appearance of the waste water, but a good share of the cleaning was done by the second scrubber, judging by the reddish-brown color of the water discharged.

73 The cleansing efficiency of the wet scrubbers, that is, the ratio of the amount of dust removed by the scrubbers to the total amount which they receive is given in Figs. 25, 27 and 28 and in Appendix 4, Table 2, showing the amount of flue dust in clean gas, with its daily and monthly variations. The average efficiency of the wet scrubbers was nearly 80 per cent during the first, and 78.8 per cent during the second half of 1909, while the average yearly efficiency reached 79.3 per cent. The drop in the second half, and particularly in September and October, is due to the ferrosilicon and spiegeleisen runs on blast furnace No. 1. If the relatively high amount of dust in the dry-cleaned gas for the same period is considered, as well as the fact that silicious dust is exceedingly difficult to remove, this reduced efficiency is not surprising.

74 Of greatest interest is the effect of wet cleaning on the amount of moisture in the gas. This information is given in Fig. 25 and Fig. 29. While the quantity of water used at the wet scrubbers was considerable, averaging 82.8 gal. per 1000 cu. ft. of gas cleaned, the aver-

age amount of moisture in clean gas was only 6.62 grains per cu. ft. with a maximum of 13.243 grains in August and a minimum of 2.61 grains in April. By comparing these figures with the corresponding average amounts of moisture in atmospheric air, an interesting coincidence will be noted, as the maxima in both cases occurred in August while the minima obtained in April. In Appendix 4 are detailed results of tests made to determine the cooling and condensing effects of the wet scrubbers.

PERFORMANCE OF SECONDARY CLEANING PLANT

75 The Theisen gas washer installation in the secondary cleaning plant was very successful, both in regard to the mechanical operation and the cleaning efficiency of the washers. Theisen washer No. 1, started in 1907, was opened for examination on February 6, 1909,

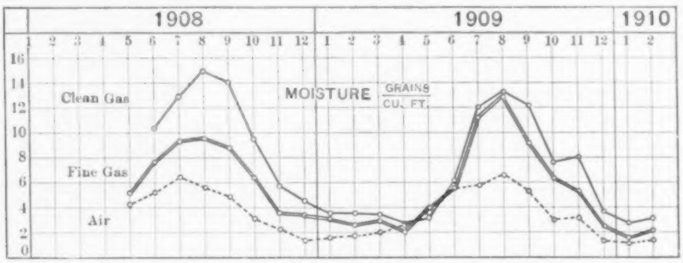


Fig. 29 Moisture in Gas and Air (Monthly Averages)

after about 7,400 hours of operation. The condition of this washer was as follows: The drum was almost perfectly clean, the longitudinal vanes showing a coating of soft mud about ½ in. in thickness and 10 in. in length on the back side near the gas inlet. The total amount of mud and dust on the vanes when dried filled a 3½-gal. water pail twice, and, except to repaint the drum, the washer needed no attention. The paint was worn off the front of the vanes only, especially on the outer edges. The wear of the water on the longitudinal vanes, due to its velocity and its contents of granulated cinders, was quite noticeable at the points were the water happened to impinge. On account of a slight construction defect and the inadvertent use of hard high-carbon steel in their manufacture some of these vanes cracked along their rivet holes and needed replacing. Softer low-carbon steel has since been used and the longitudinal vanes braced, as shown in

Fig. 19, and this trouble has not again occurred. The wear in the babbitt-lined bearings allowed the shaft to lower about 1/32 in.

76 The wire netting was absolutely clean, while somewhat corroded in places on the bottom, and the claim that this washer is self-cleaning was fully substantiated by the examination. Theisen washer No. 2 was opened in March 1910 for its second examination after nearly 9,300 operating hours, and its condition was found to be equally satisfactory. The accumulations of mud were very slight as shown in

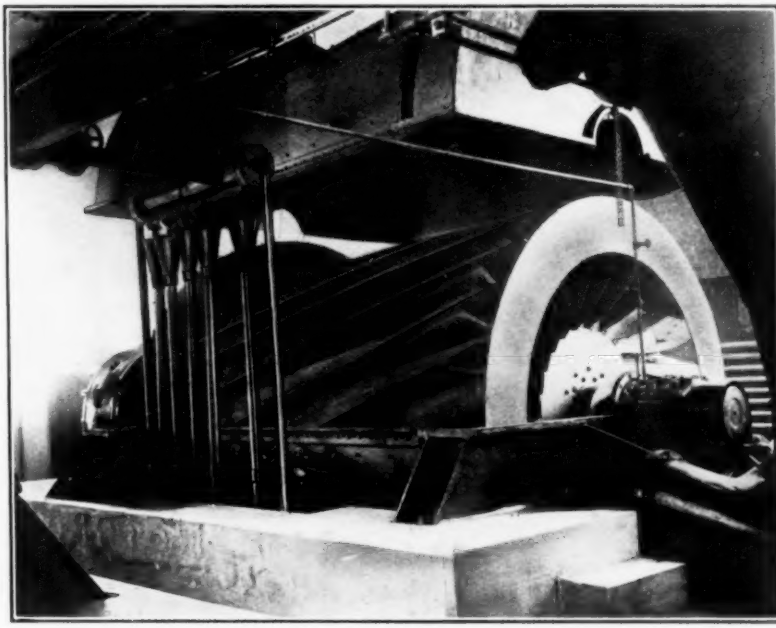


Fig. 30 Theisen Washer during Examination

photographs, Fig. 30 and Fig. 31, which were taken immediately after removing the top half of the casing. The washers can run continuously for months without being shut down, except for occasional cleaning of the motors. All smaller repairs on these washers, as well as on the whole gas-cleaning plant, are made by the operators, and the expenditure for lubricants and other supplies is amazingy small.

77 The chart in Fig. 2 shows that the average gas pressure after the Theisen washers was only slightly in excess of the aver-

age raw gas pressure, the difference being about 3 in. of water column. The advantage of this slight pressure difference is obvious, in contrast with the considerably higher pressure given by so-called hydraulic fans. Superfluous pressure for the transmission of the gas to the point of consumption, must be paid for in excess power. The gas temperature at the Theisen washer inlet was practically water temperature, while the temperature of the gas delivered by the Theisen washers was found to be on an average 2 deg. to 3 deg. higher, a difference

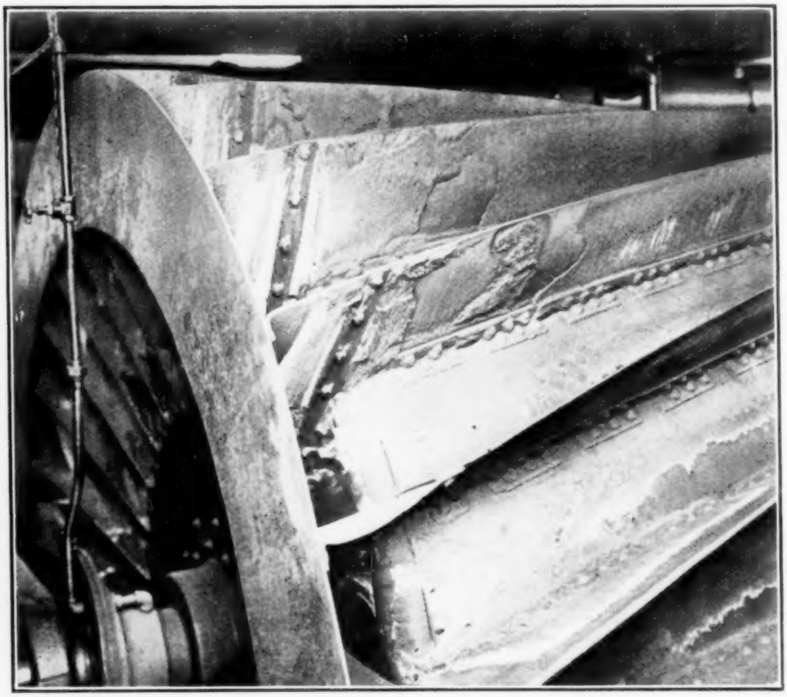


FIG. 31 THEISEN WASHER—DETAIL OF DUST DEPOSIT

explained by the fact that the mechanical work consumed by the Theisen washers must be transformed into heat, which in portion is imparted to the gas. On its way to and in the gas holder this heat is again radiated, so that the gas temperature at the engines practically equals the temperature of water supply, and only a little difference can be observed between the atmospheric temperature and the temperature of the fine gas (See Fig. 26).

78 The performance of the Theisen washer as a gas cleaner, considering the variety of conditions of furnace operation, was beyond reproach. By referring to the daily and monthly averages, given in Table 4 of Appendix 4 and plotted in the charts in Fig. 25 and Fig. 27, it will be noted that the efficiency of the secondary cleaning plant, that is, the ratio of the amount of dust removed by refining to the total amount contained in the clean gas leaving the preliminary washing plant, averaged 98.1 per cent. While this figure in itself is remarkable, it is of much more importance that this efficiency, as shown by the close coincidence of the monthly average figures, was exceedingly uniform, varying from a maximum of 99.1 per cent in October, to a minimum of 95 per cent in December 1909. The average efficiency for the first half-year was 97 per cent, while the corresponding figure for the second half reached 98.7 per cent. The great variations to which the amount of flue dust in dry cleaned gas was subjected during the year, especially however in March and April and during the period from the latter part of August until the middle of December, did not particularly affect the efficiency of the secondary cleaning plant, since in September and October, when the amount of flue dust in dry cleaned gas averaged 3 to 4 grains per cu. ft., the efficiency of the secondary washing plant shows a very marked uniformity—the average from August 25 until November 5 being 98.87 per cent—while the efficiency of the wet scrubbing plant gradually decreased from 83 per cent in July to 75 per cent in August, 69 per cent in September, and 57 per cent in October. The amount of flue dust in fine gas during these months was, of course, higher than in any previous period, but the efficiency of refining was nearly a constant maximum, irrespective of the dust conditions of raw and clean gas and of the quality of the flue dust.

79 During September and October, when furnace No. 1 made its ferrosilicon run, the color of the waste water from wet scrubbers and Theisen washers was very milky, and the dust so high in quantity, and so peculiar in quality, that considerable accumulations occurred in the clean gas main and at the inlet gate valves of the Theisen washer. The silicious flue dust seemed to set like cement, and lumps of considerable size and of great hardness had to be removed with a bar, from the inlet gate valves of the Theisen washer. Nevertheless the efficiency of the secondary washing plant during these two months was higher than during any other period.

80 In the previous paragraph the term "efficiency of the secondary cleaning plant" was used deliberately in order to indicate that in

this remarkable showing gas mains and gas holder participated. The clean and fine gas flues, connecting wet scrubbers, Theisen washers and power station gas holder, have a combined length of approximately 1000 ft., and a diameter of 5 ft. 6 in. and 5 ft. respectively. The total quantity of gas passing these mains per minute averaged 16,900 cu. ft., with a maximum of 20,300 cu. ft. in October, and a minimum of 13,600 cu. ft. in January. The velocity of the gas while traveling through these pipes did not exceed therefore 14 ft. per sec. in the clean gas main, and 17 ft. per sec. in the fine gas main. At this low velocity some impurities will undoubtedly drop out in the clean gas main, with the finely divided water which is carried along by the gas. The fine gas main, however, was always found practically clean; and while an examination of the gas holder tank has not been made, it is not believed that any great quantities of flue dust would be found in the bottom of the tank, as the dust in the fine gas is so impalpable that the fine gas burns with absolutely clear blue flame. However, in order not to credit the Theisen washers alone with removing 98 per cent of the dust from the clean gas, the efficiency is stated as embracing the secondary washing plant as a whole, or the combination of gas pipes, Theisen washers, water separators and gas holder.

81 That the Theisen washers must be credited with the bulk of the work is proven by the test summarized in Table 5 of Appendix 4. From the results given it will be seen that the efficiency of the clean gas main averaged 16.5 per cent while the fine gas main and the gas holder removed only 6 per cent of the dust delivered by the Theisen washers, or 0.23 per cent of the dust in the clean gas, proving that but little cleaning is done by gravity and reduced velocity. The Theisen washers had nearly 95.5 per cent absolute efficiency during the week when the tests were made, while their relative efficiency, based on the amount of dust in clean gas, was 79.64 per cent. It is safe to assume that similar conditions prevailed during the year 1909 and that the same relative proportions hold true at any time. The inconsistency in some of the results obtained after the Theisen washers and after the gas holder, which would indicate the impossible condition that the gas picked up a certain amount of dust on its way from washers to holder, is due to the small quantities on the shell of the Brady filter, which have to be dealt with in fine gas and cause errors in observations.

82 Since starting the gas power plant in 1907 the absolute amount of dust in the fine gas naturally continued to increase gradually, corresponding to a similar increase in the amount of gas cleaned per

minute, shown in Fig. 32. At the beginning of operations the gas engines received gas of a degree of cleanliness excessive for practical purposes. Thus the average dust contents in the fine gas in the second half of 1908 was only 0.0036 grains per cu. ft., or 0.0077 grams per cubic meter, which is much less than the average usually guaranteed. An excessive purification of blast furnace gas, even for engine purposes, is unwarranted, because the atmosphere in a steel plant is usually very dirty, and it seems quite out of place to purify the gas to a higher degree of cleanliness than the combustion air, unless the latter is to be subjected to a similar cleaning process. Tests made in July 1909 to determine the quantity of impurities contained in the air near the air intake of the gas engines showed the amount to be between 0.0005 and 0.0052 grains per cu. ft. with an average of determination (given in

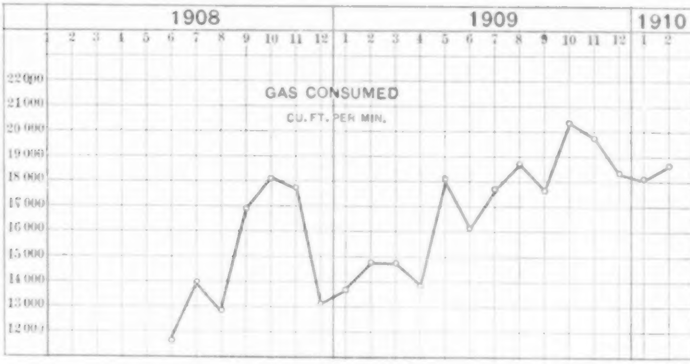


FIG. 32 GAS CONSUMED, CU. FT. PER MIN. (MONTHLY AVERAGES)

Table 6, Appendix 4) of 0.00346 grains per cu. ft. The amount of dirt in the air is therefore by no means a negligible quantity. The detrimental character of these impurities is also apt to be under-estimated. In Appendix 4, Table 7 is a comparison of the chemical analyses of samples of the deposit on the gas and air dampers in one of the gas engines, taken in February 1909. It is to be noted that the gas never comes in contact with the air dampers, so that the samples fairly represent the accumulation of dirt deposited by the combustion air.

83 A comparison of the four analyses shows the surprising fact that the amount of iron in the dust sample taken from the air dampers is about twice as great as in the dust deposited on the gas dampers. If this is true—and there is no room for doubt, as the samples represent the accumulations of more than one year, the dampers never having

been cleaned—it follows that appreciable quantities of iron, sand and coke enter the engines with the combustion air.

QUALITY OF FLUE DUST

84 The chemical composition of the flue dust removed from the gas at various stages of the cleaning process was made the subject of analysis in March 1908. Samples were taken at the following points: dry deposit from the main water seal and collecting flue after dry dust catchers, suspended matter secured by evaporation of samples of the waste water from wet scrubbers No. 1 and No. 2 and the Theisen washers. The results of this test are shown in Table 8, Appendix 4, all analyses giving metallic iron and manganese as Fe and Mn, while the constituents exist in the form of oxides. Fixed carbon is mostly coke, and the volatile is the CO_2 from limestone. This table shows that the relative amounts of silica, alumina, lime, etc., increase gradually, more and more of the heavy impurities such as Fe dropping out the further the cleaning process progresses.

85 A similar test was made in March 1909, with the difference, however, that an attempt was made to determine the quality of the dust remaining in the gas while passing the various stages of gas cleaning. In this case the method of securing samples consisted in connecting Brady filters at the various points and turning the gas on simultaneously at all filters. Irrespective of the size of the gas sample, the cartridges were not removed until sufficient quantities of dust had collected for quantitative and qualitative analyses. It was impossible to obtain a sufficiently large sample for analysis, on the Brady filter installed after the Theisen washers, for while the filter passed the gas very rapidly for a few hours, the flow gradually diminished as the dust in the fine gas closed up the pores of the paper. After a 36-hour run the flow through the filter had stopped completely and the dust collected on the Soxhlet tube appeared as a dull gray coating, which could not be removed. The results of this experiment are given in Table 9, Appendix 4.

86 The relative amount of SiO_2 carried in the gas is practically constant before any wet washing takes place, but the percentage increases in the clean gas, with decreasing relative contents of iron. The wet scrubbers remove the bulk of the iron dust, while the lighter impurities are carried over into the Theisen washers. A sample of the dust deposited in the gas pipe on one of the gas engines immedi-

ately before the gas enters the cylinders at the end of its travel, taken March 15, 1910, was analyzed as follows:

SiO_2	Al_2O_3	Fe	CaO	MgO	Mn	Vol.
36.20	7.53	7.18	12.50	0.90	0.49	34.32

The high percentage of silica, lime and volatile matter, and the very low contents in iron, are noteworthy, and show that silicious dust, lime and coke are carried by the gas much farther than iron, which is removed to the greatest extent in the first stages of cleaning.

87 Due to sulphur in the coke, blast furnace gas contains a certain amount of this impurity in the form of H_2S and SO_2. Its pres-

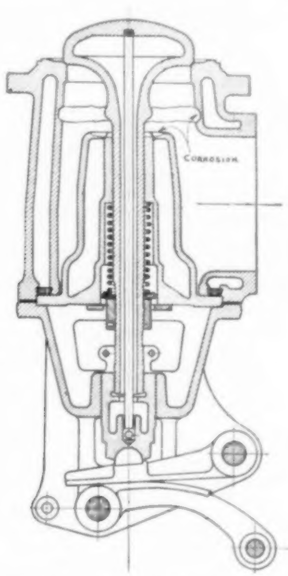


FIG. 33 CORROSION OF VALVE CASING BY SULPHUR IN GAS

ence was discovered after some length of operation of the gas engines. It was found that several of the exhaust valve casings showed very pronounced corrosion, at a place indicated in Fig. 33 and it was concluded that some chemical action had taken place. Two tests made Feb. 2, 1909, showed the presence of 0.0416 and 0.0407 grains per cu. ft. of sulphur in the gas. It was further discovered that the fine flue dust accumulating in thin layers in the gas passages to the engine cylinders, can be lighted and glows with a blue color, unmistakably giving out SO_2 vapors. The corrosion in the exhaust valve chambers was caused by water leaks from the original inner exhaust valve guides,

which cracked as indicated in the illustration. The water coming in contact with the hot exhaust gases formed H_2SO_4, which impinging on the wall of the exhaust valve chamber, had the corrosive effect. The presence of sulphur in blast furnace gas for this reason prohibits the use of sheet-iron exhaust pipes and mufflers, while corrosion has never been observed when cast iron is used. It was attempted, some time ago, to utilize the waste heat from the exhaust of these engines to raise low-pressure steam for use in the heating system of the power house, in wrought-iron pipe coils arranged inside the cast iron mufflers, and while the results were very satisfactory from the standpoint of heat transmission, this heating system was a failure on account of the rapid corrosion to which the heating coils in the mufflers were subjected.

88 The amount of moisture remaining in clean gas, fine gas and air, as well as the variation from day to day and from month to month, are shown in Figs. 25 and 29.

89 While the Theisen gas washers receive the gas at practically water temperature, so that a condensation of water vapors by cooling is improbable, the amount of moisture in the engine gas is nevertheless lower than in the clean gas, and this in spite of the exceedingly intimate contact between gas and washing water in the Theisen washers. The indications are therefore that the Theisen washers remove not only dust, but moisture as well, probably by the action of centrifugal force, which throws gas and water vapors against the circulating water film on the inside of the stationary casing, thereby drying the gas mechanically. Furthermore a great deal of moisture is being deposited in the fine gas main and gas holder. The average moisture in the gas delivered to the engines was 3.39 grains for the first half, and 7.85 grains for the second half of 1909, with a yearly average of 5.62 grains per cu. ft. Comparing these figures with the corresponding values in clean gas and atmospheric air, it will be seen that the average moisture in the engine gas for the year is 60 per cent higher than the average moisture in the atmosphere, which is considered very favorable in view of the high moisture contents in raw gas and the large quantity of water which is brought into such intimate contact with the gas.

90 The effect of the sun beating down on gas mains and gas holder is very noticeable in the summer months, accounting for the high moisture contents in July, August and September. Pipes and holder become quite hot and the finely divided mist in the gas is rapidly evaporated, as indicated by a drip installed near the venturi meter.

Drops or small streams of water are freely discharged in the winter months, while the dripping ceases as soon as the atmospheric temperature exceeds about 78 deg. fahr. The moisture carried with the gas into the engine cylinders has not given cause for trouble at any time.

WATER AND POWER CONSUMPTION OF CLEANING PLANT

91 Fig. 34 shows the average monthly consumption of water since July 1908 in gallons per thousand cubic feet of gas cleaned, as

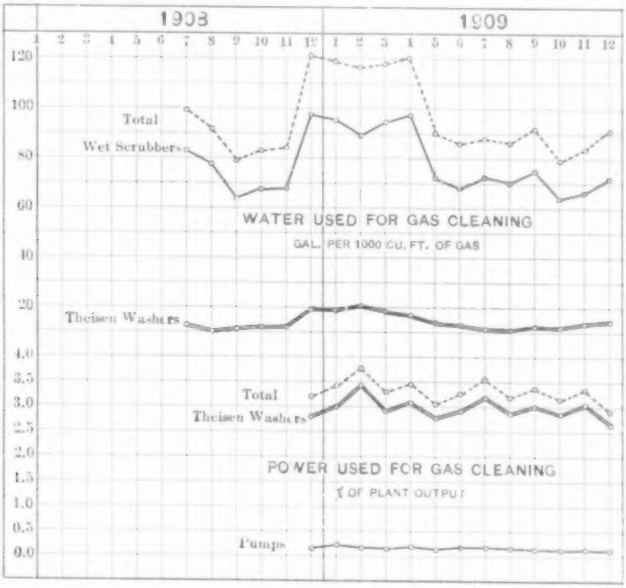


FIG. 34 WATER AND POWER CONSUMPTION OF GAS CLEANING PLANT

established from daily weir measurements. It is to be noted that the water consumption of the wet scrubbers is about four times as high as that of the Theisen washers, for the same quantity of gas cleaned. The latter varied from 26.1 gal. in February to 16.0 gal. in July, with an average of 21.8 gal. for the first and 17.0 gal. for the second half, and 19.4 gal. per 1000 cu. ft. of gas for the whole year. The corresponding figures for the wet scrubbers were a maximum of 103.2 gal. in April, and a minimum of 68.6 gal. in October, with an average of 91.0 gal. for the first half, 74.6 gal. for the second half, and 82.8 gal. per 1000 cu. ft. for the year 1909.

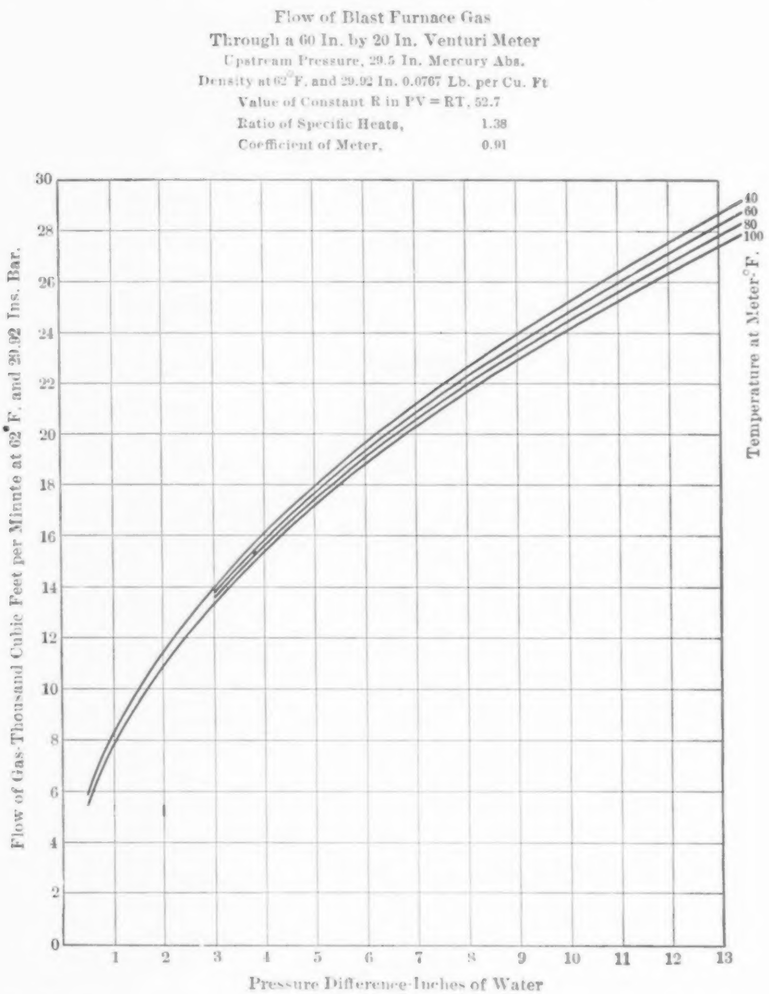


FIG. 35 VENTURI METER CURVES SHOWING FLOW OF GAS FOR DIFFERENT METER READINGS

92 The high water consumption in the first part of the year is due to the smaller quantity of gas cleaned. About the same absolute quantity of water was maintained on the scrubbers. Generally speaking, the quantity of water used in the wet scrubbers could probably be reduced without impairing their efficiency, but it is preferred to use water in excess rather than to run the risk of clogging the sewers, especially since the question of economizing washing water is of no particular importance in a plant located on the lake front.

93 In regard to the amount of power required by the gas cleaning plant, Fig. 34 shows the monthly average power consumption of wet scrubber pumps and Theisen washers, expressed in per cent of the output produced by the gas engines. About 90 per cent of the total power is being used by the Theisen washers, only 10 per cent being

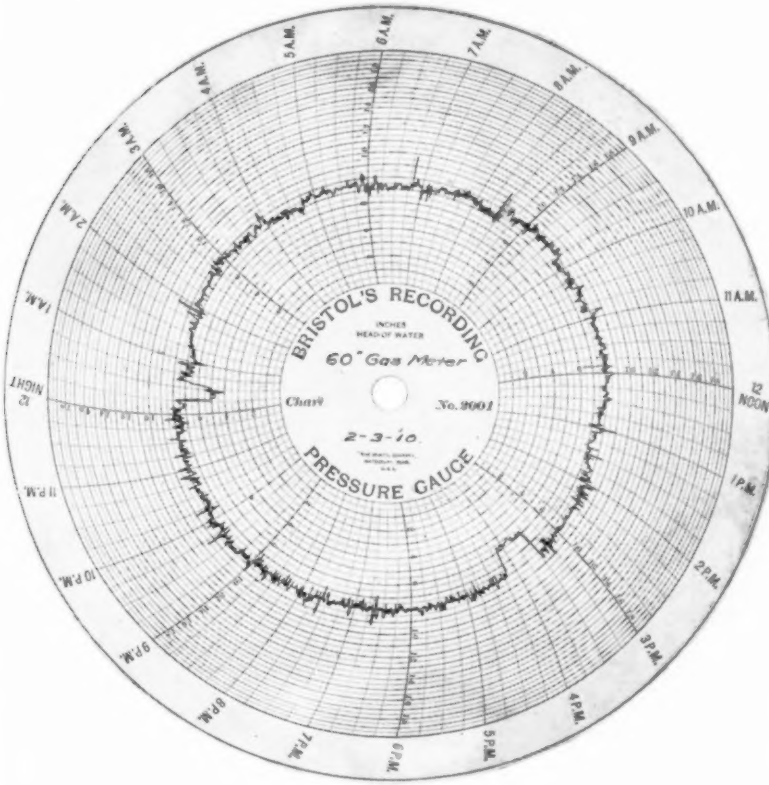


FIG. 36 BRISTOL CHART OF PRESSURE DIFFERENCES, 60-IN. VENTURI METER

necessary to operate the wet scrubber pumps. The average power consumption of the Theisen washers was 2.977 per cent of the total output of the station, and the respective values for the first and second halves of 1909 were 3.00 per cent and 2.931 per cent, with a maximum of 3.44 per cent in February and a minimum of 2.649 per cent in December.

94 These figures are somewhat higher than are often claimed for similar washers abroad, but an average power consumption for the gas-cleaning plant, of from 3 per cent to 3.5 per cent of the total power output of the gas engines, should not be considered excessive in view

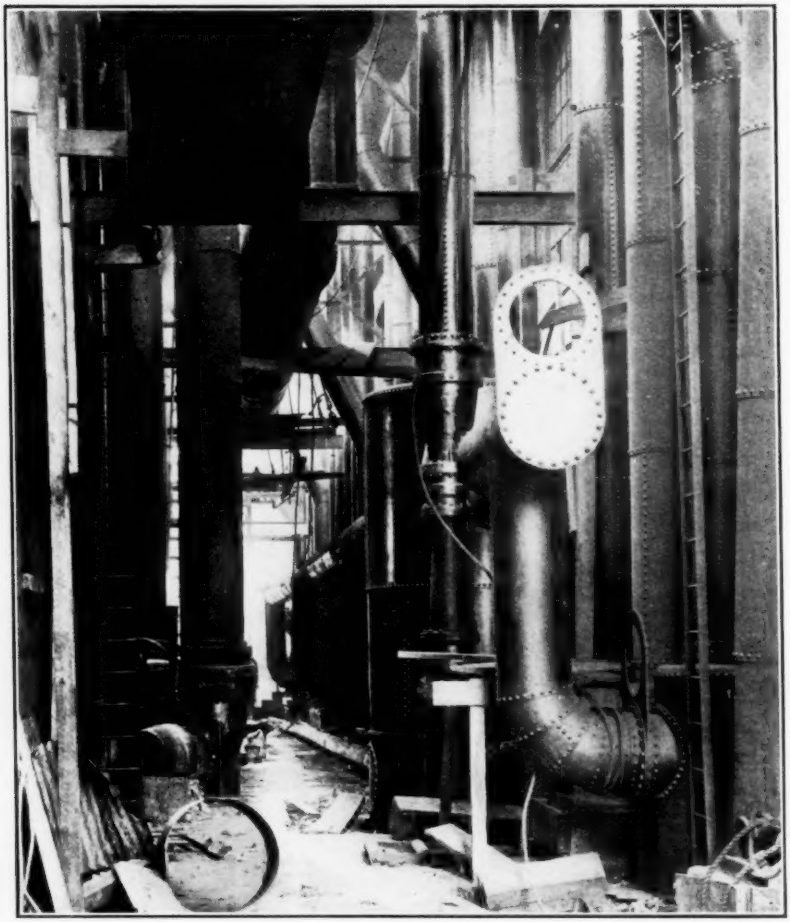


Fig. 37 Exterior View of Test Piping

of the benefit which is being derived from this expenditure. It is worthy of note also that the engine builders who furnished the gas engines for the plant under discussion have never found fault with the physical condition of the gas, while more or less vigorous com-

plaints about "excessive dirt" in the gas are generally advanced as soon as trouble develops in the engines. The necessity of auxiliary machinery is furthermore not characteristic of gas engine installations alone, as boiler-feed pumps, hot-well pumps, dry-air pumps, forced-draft ventilators, mechanical stokers, etc., are indispensable for steam engine plants consuming probably as large a percentage of the power developed.

95 It has been shown in the previous pages that the blast furnace gas was delivered to the gas engines in a highly satisfactory physical condition. The gratifying results of the thorough cleaning and refining of the gas were, that no difficulties which might have been caused by insufficiently cleaned gas were ever experienced in the operation of the gas engines, and these engines never had to be stopped for the specific purpose of cleaning internally the gas valves and gas passages. The amount of dust deposited on internal engine parts was invariably so small that it could be brushed off with the finger, and these engines could undoubtedly operate at full-load capacity a whole year and longer without cleaning the gas inlet passages and cylinders.

96 Blast furnace gas delivered to the power house is charged to operation of the engines at a value based on the price of coal with the cost of cleaning and refining added to the value of the raw gas, which is established on the basis of equivalent heat values. In order to determine the charge made for purified blast furnace gas delivered to the gas power plant, a continuous record is being kept of the quantity of gas blowing to the gas holder, venturi meters being used as measuring instruments. Details of the methods are given in Appendix No. 5. Charts relating to meter measurements are shown in Figs. 35 and 36 of the paper.

THERMAL EFFICIENCY AND OUTPUT OF GAS ENGINES

97 It seemed desirable in connection with the installation of six gas blowing-engines, furnished by three different manufacturers, to provide means for determining the thermal efficiency of each type of engines without interfering with the regular operation of the others. To this end a special venturi meter was installed and connected to a "test" pipe, shown in Figs. 23*a* and 37. By opening the disc valve located inside the overhead gas receiver, which controls the flow of gas through the venturi meter into the large reservoir for equalizing pulsations caused by the intermittent suction strokes, and test piping, and by turning one small spectacle valve while simultaneously

shutting off the direct gas supply from the overhead receiver by filling the water seal, any engine can be operated on measured gas while the others receive gas directly through the branch pipes. Such tests can be commenced at any convenient time, and prolonged and repeated at will. A Bristol differential pressure recorder installed in the blowing engine-room gives continuous records of each test, so that reliable averages can be obtained of the gas consumption under different operating conditions.

98 Fig. 38 shows the monthly averages of the kilowatt output of the power station, of the heat consumption in B.t.u. per kw-hr. and per b.h.p.-hr. based on a generator efficiency of 96.2 per cent at full load, and of the thermal efficiency. The diagram also shows the total operating time of each engine since starting.

99 The thermal efficiency of the plant was very uniform from the middle of 1908 until about May 1909, averaging 23.22 per cent. The drop which began in May and reached a low value in October, was due to certain troubles encountered with gas cylinders and piston rings, etc. These could not be remedied at that time, as on account of the ever-increasing demand of electric power and the lack of a spare unit, it was impossible to shut the gas engines down sufficiently long for a thorough overhauling and for necessary repairs. Doubtless with an additional spare engine the load factor would have been lower than the average for 1909, which reached 72 per cent, but the thermal efficiency would have remained constant—or nearly so—since the necessary repairs, adjustments and changes could have been made on these engines in time, without reduction in the total kilowatt output of the power plant. In spite of this reduced efficiency in the second half of 1909, the average figure obtained for the whole year,—not as the result of one or of several individual tests, but as the fair average of daily observations, proper corrections having been made for the inexact readings caused by dust in the venturi meter in September and October—was 20.8 per cent, with a maximum monthly average of 23.77 per cent in March, and a minimum of 17.8 per cent in October. The highest daily average efficiency in 1909 was 25.7 per cent on March 11.

100 During the year 1909, the engines, while in operation, ran at nearly full-load capacity, the average for the four engines ranging from 93.60 to 99.63 per cent.

101 The values of the total kilowatt output of the gas power plant for each month since regular operation was begun, are plotted in Fig. 39, showing that the maximum output for any month occurred in

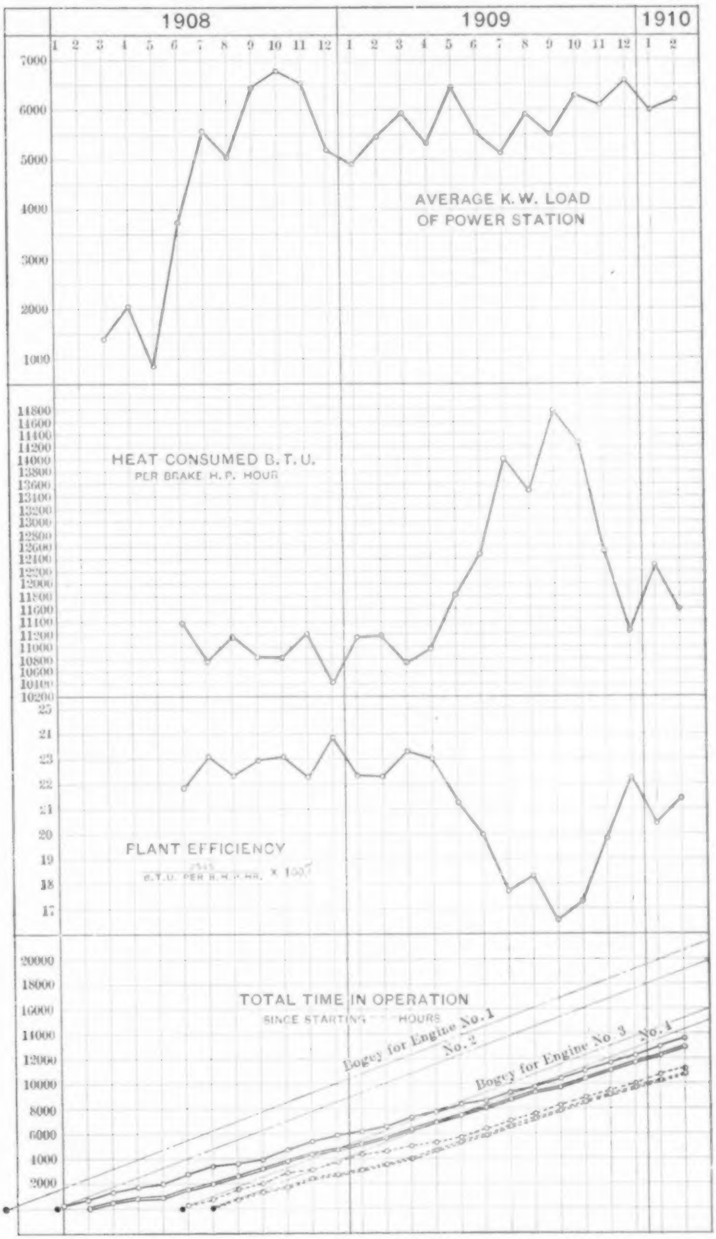


FIG. 38 AVERAGE LOAD, HEAT CONSUMPTION, PLANT EFFICIENCY AND OPERATING TIME (MONTHLY AVERAGES)

October 1908, when 5,000,000 kw-hr. was produced. The output of the gas power plant for the year 1909 was 50,494,100 kw-hr. against 43,953,640 kw-hr. produced by the steam-driven generators. The load factor for 1909 of the gas power plant was 72 per cent, against

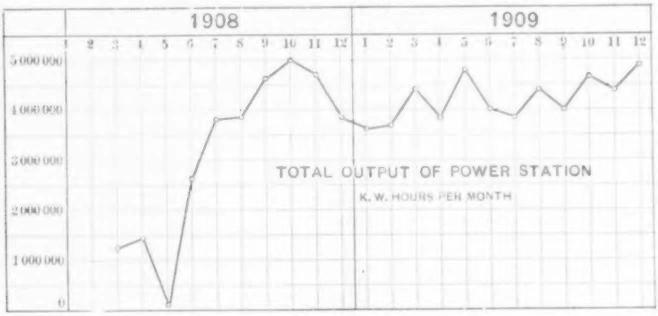


FIG. 39 TOTAL OUTPUT OF GAS POWER PLANTS (MONTHLY AVERAGES)

47 per cent for the steam power stations, which have a rated capacity of 10,900 kw. The total output of electric power generated in 1909 was 94,447,740 kw-hr., 53.5 per cent of which was produced by the gas power plant.

APPENDIX NO. 1

MONTHLY RECORDS OF THE POWER PLANT (8000 KW.)

TABLE 1 MONTHLY RECORD OF KILOWATTS PRODUCED PER HOUR

Month 1909	Kw. per Hr.	Per Cent of Capacity	No. of Furnaces in Blast
January	4920	61.5	3
February	5450	68.0	3
March	5940	74.0	2
April	5330	66.5	2
May	6440	80.5	3
June	5550	69.0	4
Average for first half	5600	70.0	
July	5140	64.0	5
August	5930	74.0	5
September	5500	68.5	6
October	6270	78.0	6
November	6080	76.0	5
December	6600	82.5	5
Average for second half	5920	74.0	
Average for 1909	5760	72.0	

TABLE 2 MONTHLY RECORD OF OPERATING TIME

1909	Plant in Operation		Plant down			
			Due to engines		Due to outside causes	
Month	Hrs.	Per Cent	Hrs.	Per Cent	Hrs.	Per Cent
January	426	57	213	28.5	105	14.5
February	456	68	85	12.5	131	19.5
March	529	71	97	13.0	118	16.0
April	467	65	170	23.5	83	11.5
May	609	82	73	9.5	62	8.5
June	556	77	97	13.5	67	9.5
Average for first half	507	70	123	17.0	94	13.0
July	607	82	103	14.0	36	4.0
August	594	80	131	17.5	19	2.5
September	595	83	107	14.5	18	2.5
October	668	90	52	7.0	24	3.0
November	632	88	65	9.0	23	3.0
December	609	82	52	7.0	83	11.0
Average for second half	617	84	85	11.4	34	4.6
Average for 1909	562	77	104	14.2	64	8.8

TABLE 3 MONTHLY RECORD OF LOSSES DUE TO OUTSIDE CAUSES

Month 1909	Plant down			
	Due to operation		Due to lack of gas	
	Hrs.	Per Cent	Hrs.	Per Cent
January	6	5.5	99	94.5
February	43	33.0	88	67.0
March	36	30.0	82	70.0
April	17	20.0	66	80.0
May	61	98.5	1	1.5
June	59	88.0	8	12.0
Average for first half	37	39.5	57	60.5
July	36	100.0	..	....
August	19	100.0	..	....
September	15	83.5	3	16.5
October	24	100.0	..	....
November	23	100.0	..	....
December	78	94.0	5	6.0
Average for second half	33	97.0	1	3.0
Average for 1909	35	55.0	29	45.0

APPENDIX NO. 2

DATA UPON GAS PRODUCED IN THE BLAST FURNACES

Methods of Calculation of Quantity of Gas Produced

The "nitrogen method" assumes that the nitrogen in the air, amounting to 79.3 per cent by volume, passes through the blast furnace unchanged, so that the same quantity must be found in the exit gas of the furnaces. Since the average gas composition is known, the percentage of nitrogen can be determined by difference. The amount of air blown is based upon the revolutions of the blowing engines corrected for volumetric efficiency of the blowing tubs and for 5 per cent assumed loss of air between tubs and tuyeres. Thus, for instance, in August 1909 blast furnace No. 6 produced on natural blast a daily average of 496.5 tons of Bessemer iron, using 37.5 per cent Pocahontas and 26.5 per cent Connellsville coke with an average coke consumption of 2148 lb. per ton of iron. The average gas analysis for the same month was as follows:

CO_2	CO	H	CH_4	N (by difference)
14.23	25.28	4.65	0.23	55.61

B.t.u. per cu. ft. at 62 deg. fahr. = 96.8
B.t.u. per cu. ft. including sensible heat at 500 deg. = 105.3
Temperature of air at blowing engines = 94 deg. fahr.
Cu. ft. of air blown per minute = 40,990
Cu. ft. of air blown per minute at 62 deg. fahr. = 38,610
Average blast pressure = 15.1 lb.
Cu. ft. of air to furnace per minute including 5 per cent loss at 62 deg. fahr. = 36,690
Volume of gas per volume of air at 62 deg. N unchanged = 1,426
Cu. ft. of gas per minute N method = 52,300

According to this calculation No. 6 furnace produced about 153,000 cu. ft. of gas per ton of pig iron made in 24 hours.

2 The nitrogen method, however, is subject to serious errors from the presence of moisture and foreign gases in the air as blown, and from air leakage and inefficiency of tubs. The "carbon method," which is considered more reliable, assumes that the carbon entering the furnace in the form of coke, must reappear in the form of gas, the amount of carbon in the limestone being equal approximately to that in the iron and slag. If A = per cent of carbon in the coke as charged, B = pounds of carbon in 1 cu. ft. of blast furnace gas, the amount of gas produced in cubic feet per minute = $\frac{\text{tons of iron per day}}{1440} \times \frac{\text{coke rate}}{B} \times A$.

Calculating the amount of gas according to this method, furnace No. 6 liberated 51,310 cu. ft. of gas per min., or 149,000 cu. ft. per ton of product in 24 hours.

TABLE 1 DISTRIBUTION OF GAS FROM BLAST FURNACE NO. 6

AUGUST 1909

	Million B.t.u.	Per Cent
Total gas generated	324.1	100
Stoves and leakage	130.0	40
Blowing engines	92.1	28.4
Used at furnace	9.0	2.8
Auxiliaries	4.6	1.4
Total used for blast furnace operation	235.7	72.6
B.t.u. surplus for furnace	88.4	27.4

B.h.p. equivalent of surplus 1470

3 The results of nitrogen and carbon methods, do not, however, always agree as closely as in this instance. The average coke analysis for the month of August, from which the amount of carbon charged into the furnace was determined, was as follows:

	Per cent moisture	Per cent fixed carbon (dry)
Pocahontas	2.64	90.10
Connellsville	1.94	87.38

4 For the distribution of gas it is assumed that 40 per cent goes to the stoves and is lost by leakage. The amount of gas used for the blowing engines is calculated from the boiler horse power, the latter being determined by the quantity and pressure of air blown, the steam rates of engines and the evaporation per boiler horsepower. The B.t.u. equivalent of boiler horsepower for steam distribution in the current month is equal to $\frac{\text{boiler h. p.} \times 33{,}320}{\text{boiler efficiency}}$ and the latter is arbitrarily assumed to be 55 per cent. It is further assumed that 150 boiler h.p. is used in steam at each blast furnace and that all auxiliaries use 5 per cent of the steam going to the blowing engines. The difference is the surplus gas per furnace available for other departments, which is stated in the equivalent of boiler horsepower. Using the former example for an illustration, the total amount of B.t.u. in the gas produced per hour by furnace No. 6 was in August 1909, according to the carbon method, 324.1 million. The distribution was as above in Table 1.

5 The surplus gas was used for operating gas engines in the gas electric station and for raising steam for steam electric power. In this way a B.t.u. balance can be made for all furnaces in each month, as the total kilowatt hours produced in the electric stations and the amount of coal which had to be fired under the boilers are known.

TABLES RELATIVE TO BLAST FURNACE GAS

TABLE 2 GAS PRESSURES

MONTHLY AVERAGES, INCHES OF WATER

1909	Jan.	Feb.	March	April	May	June	Jan.-June Average	
Entering gas cleaning plant..	5.0	5.7	7.3	7.6	7.1	7.6	6.7	
After Theisen washers	8.0	8.0	10.1	9.4	9.7	9.8	9.2	
Barometer, inches of mercury	29.44	29.24	29.21	29.36	29.31	29.40	29.33	
	July	Aug.	Sept.	Oct.	Nov.	Dec.	July-Dec.	Jan.-Dec.
Entering gas cleaning plant	9.1	10.0	12.0	13.4	14.6	12.2	11.9	9.29
After Theisen washers.........	11.9	15.0	13.9	16.5	17.7	14.8	14.9	12.10
Barometer, inches of mercury......	29.34	29.37	29.44	29.45	29.42	29.41	29.45	29.37

TABLE 3 COMPOSITION OF GAS FROM INDIVIDUAL FURNACES

AVERAGES

Blast Furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.	Product
1	4.36	33.71	3.41	7.75	120.4	Ferro-Silicon
2	13.47	26.34	4.43	1.95	96.7	Basic
3	14.98	23.97	4.43	1.60	91.4	Basic
4	13.91	24.99	4.10	1.79	94.1	Basic
5	14.17	25.61	3.85	1.81	95.5	Bessemer
6	13.65	25.32	4.26	1.85	95.7	Bessemer

TABLE 4 COMPOSITION OF MIXTURES OF GAS FROM VARIOUS FURNACES

AVERAGES

Blast Furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.	Product
1 and 2..............	8.92	30.02	3.92	4.85	108.5	....
1, 2 and 3..........	10.60	28.01	4.09	2.65	102.8	....
2, 3 and 4...........	14.12	25.10	4.32	1.80	94.1	Basic
1, 2, 3 and 4........	11.68	27.25	4.09	2.33	100.6	....
2 and 3..............	14.23	25.15	4.43	1.81	94.0	Basic
1, 2, 3, 4, 5 and 6...	12.25	26.66	4.08	2.23	98.9	....

TABLE 5 AVERAGE COMPOSITION OF BLAST FURNACE GAS

AT 62 DEG. FAHR. AND 30 INCHES MERCURY

	Jan.	Feb.	Mar.	Apr.	May	June	Avg. Jan.-June
CO_2	13.26	13.38	11.53	12.43	13.10	13.20	12.82
CO	25.61	25.50	28.10	26.67	26.56	26.50	26.49
H	2.99	3.95	2.92	3.16	3.74	3.89	3.44
CH_4	0.21	0.23	0.24	0.21	0.22	0.18	0.215
Computed B.t.u	93.04	95.70	100.50	98.81	97.70	98.20	97.32
Ratio $\frac{CO}{CO_2}$	1.93	1.90	2.43	2.15	2.02	2.00	2.07
Heat value per cu. ft. by Calorimeter B.t.u	93.45	96.10	101.00	98.08	98.32	97.99	97.49

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
CO_2	14.10	12.50	10.03	11.96	13.88	13.75	12.53	12.67
CO	25.90	27.30	29.80	26.02	24.70	25.85	26.54	26.51
H	3.86	4.06	3.77	3.45	3.59	3.98	3.78	3.57
CH_4	0.17	0.18	0.19	0.21	0.19	0.15	0.18	0.196
Computed B.t.u	95.70	101.40	108.70	102.90	86.70	95.80	98.40	97.90
Ratio $\frac{CO}{CO_2}$	1.83	2.18	2.98	2.17	1.78	1.88	2.12	2.09
Heat value per cu. ft. by Calorimeter B.t.u	95.50	101.60	107.40	103.10	92.90	94.60	99.20	98.30

APPENDIX NO. 3

DESCRIPTION OF METHODS AND INSTRUMENTS USED IN OBTAINING DATA UPON THE PERFORMANCE OF THE GAS CLEANING PLANT

Temperatures. The temperature of the gas entering the cleaning plant is measured and automatically recorded by a Bristol pyrometer. Readings of the gas temperature are taken by the operators every three hours, after the dry cleaning plant, after the wet scrubbers, and before and after the Theisen washers, the last temperature being also recorded by a Bristol recording thermometer. Temperature readings are further taken at the inlet and outlet of each gas holder. All thermometers have the Fahrenheit scale and are permanently installed in the pipe lines. The water temperatures are read every three hours at the pump suction, after each wet scrubber and after the Theisen washers. For the purpose of comparison the temperature of the atmosphere near the washer building is simultaneously recorded. Portable thermometers with the Fahrenheit scale are used for measuring water temperatures.

2 *Pressures.* Readings are taken by the operators every three hours, by means of ordinary U-tubes, of the gas pressure in inches of water at the following places: at the point where the gas enters the cleaning plant, between the *wet* scrubbers, and before and after the Theisen washers. In addition, the pressure of the raw gas at the main water seal, and of the fine gas after the Theisen washers, is continuously recorded by Bristol pressure gages. For convenience of observation all instruments indicating and recording gas and water pressures, temperatures and venturi meter pressure differences, are arranged on a *gage* board (Fig. 1)installed in the Theisen washer building. On the same gage-board the telephone gongs and the optical indicators showing the position of the two gas-holder bells are mounted. By a simple system of contacts each gas-holder bell, while descending, closes five different electric circuits, and causes incandescent lights to burn corresponding to its different positions. Thus a white light burns when the holder is in its top position; a green light appears when the holder bell has descended 7 ft.; one red light, indicating danger, corresponds to a 14-ft. immersion of the holder bell, and two, and at last three, red lamps show that the bell has fallen 21 ft. and 28 ft. respectively, and is nearing its bottom position. When the three red lights burn the gas holder is practically empty. Similar optical indicators are installed in the electric power and gas blowing-engine houses, so that the engine operators can independently observe the position of the gas holders at any time.

3 *Power Consumption.* As all Theisen washer and pump motors are operated by 440-volts alternating current reduced in special transfomers from 2200 volt, 25 cycle, 3 phase current generated in the electric station, an integrating kilo-

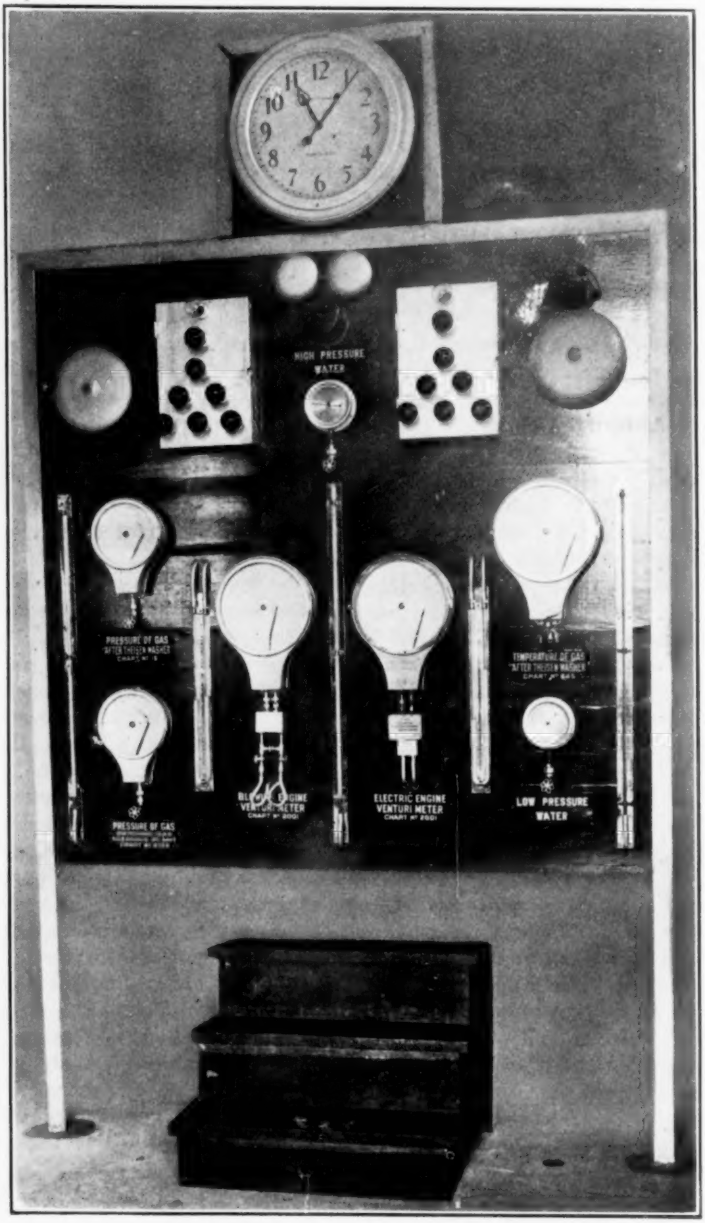


Fig. 1 Gageboard in Theisen Washer Building.

watt meter was installed at the main switchboard to measure the combined power consumed by all gas cleaning plant motors. Unfortunately, the motor-driven air compressors for starting the gas engines are on the same line, so that the power for two 50-h.p. motors is included in the meter readings; but since each air compressor is running only about one hour each day, the error is believed to be of little weight. However, it must be kept in mind that on this account the recorded power consumption of pumps and Theisen washers is higher than the actual values. A portable ampere meter for use in the Theisen washer building can easily be attached to each motor and read by the operator every three hours, and the readings are recorded in the daily report sheets. These ampere meter readings give the only indication of the load carried by each Theisen washer. As all washers operate in parallel, one machine, by a slight misadjustment of the inlet and outlet gate valves, may handle more gas, thus carrying more load, and cleaning its share of the gas to a less extent than the other washers. The ampere meter will indicate such an inequality in the load distribution, and the operators have orders to keep the ampere readings on all washers in operation fairly uniform, by proper adjustment of the gate valves. The indications of the ampere meter, averaged for each month, are used to divide the power charges in proper proportion between the pumps in the preliminary washing plant and the Theisen washers.

4 *Water Consumption.* The amount of water consumed each minute in the gas-washing plant is measured by overflow weirs at the settling tank. A hook gage gives directly the number of gallons of water per minute falling over the weir so that the operator reads the water quantity as easily as a thermometer. The settling tank has two compartments, so that the water from the wet scrubbers can be turned into one, while the Theisen waste water is flowing into the other compartment. Gates make it possible to reverse these flows and to shut off each settling tank for cleaning. The amount of water allowed at each wet scrubber and Theisen washer, originally apportioned by means of calibrated barrels, is adjusted in practice by estimating the relative quantity basing the estimate on the thickness of the stream at each Theisen water inlet and at the wet scrubber overflow. The water used for gas-washing purposes is waste cooling water from the blast furnaces, formerly discharged into the sewer but now collected and piped to the Theisen washers under its natural head, while another part is lifted on top of the wet scrubbers by two centrifugal pumps of 2,000,000-gal. capacity each. These pumps receive the water under 30 ft. head and deliver it at a pressure of 80 ft. for distribution through the sprinkler system.

5 *Dust and Moisture.* Gas engine builders usually specify the amounts of dust and moisture which should not be exceeded for safe operation. Recent specifications call for blast furnace gas containing not more than 0.02 grains of flue dust and not over 10 grains of moisture per cu. ft., at 62 deg. fahr. and 30 in. mercury. With an efficient, modern gas-cleaning plant it is not difficult to meet these conditions, as is shown in the paper. Dust and moisture determinations are made in the gas laboratory, and recorded on daily chemical report blanks (Fig. 6 in the body of the paper). The amount of dust is determined in dry cleaned gas, clean gas and fine gas, and occasionally in the atmosphere, while moisture determinations are made in clean and in fine gas. For purposes of comparison the moisture in the atmosphere, as determined at the dry-blast

plant, is also recorded. Results are given in grains per cubic foot of standard gas, and by a simple calculation the efficiency of the wet scrubbers and of the secondary washing plant can be determined each day.

6 While frequently tried, it has been found impossible to make dust determinations in raw gas, which give more than a general idea of the efficiency of the dry cleaning plant. The difficulty is that the raw gas frequently carries larger particles of dirt, which obstruct the sample pipe and clog the dust filter before a gas sample of sufficient size for correct determination can be secured.

7 A series of tests extending over eight days was made in August 1908, with the regular dust filter apparatus then in use, but since no results were obtainable, a long glass tube open at both ends and filled with dry calcium chloride was then weighed and attached to the gas main in place of the dust filter. After passing several cubic feet of gas, the glass tube was taken off, closed at both ends and weighed, the difference in weight giving the total amount of dust and moisture. The amount of dust was determined by drying and weighing again as the difference between the second and third weighings. This apparatus was found to be a little more satisfactory, as larger samples could be taken, but gradual filling of the pipe with dust made the accuracy of the results very doubtful. The size of the gas sample secured never exceeded 6 cu. ft., with an average of about 2 cu. ft.

8 Tests made in August 1907 gave similarly unreliable results, as the amount of dust in raw gas, according to these determinations, varied from 0.18 grains per cu.ft.at 2 p.m., August 7, to 563.19 grains per cu. ft.at 4.30 p.m. August 29. Such extreme variations are improbable, and as the results obtained cannot represent a fair average, raw gas dust tests were discontinued as of questionable value.

9 The method and the instrument used for the determination of dust in clean and fine gas were developed by Messrs. Wm. Brady and L. A. Touzalin. The Brady filter (Fig. 2) consists essentially of a brass cylinder provided with inlet and outlet and supporting the filter itself, which is an ordinary 94 x 33 mm. Soxhlet extraction shell. The brass cylinder is of such diameter that an annular space of about $\frac{3}{16}$ in. is formed between its inner surface and the paper filter. The filtering shell is fastened and held tightly in place without the aid of gaskets, by wedging its open end between the tapering cylindrical faces of the brass shell and the brass nozzle, as shown in the illustration. This method of fastening has the additional advantage that for a distance of about one-half an inch from the edge, the Soxhlet shell is protected from dust deposits so that the filter can safely be handled after the experiment. The brass nozzle forming the inlet of the apparatus is provided with inside threads so that it can be screwed to a sampling pipe. Its inside surface is perfectly smooth, without ledges or places for the accumulation of dust. A brass nut holds the three parts of the instrument in place. The apparatus may be used in any position, but the preferred arrangement is horizontal.

10 When the gas to be filtered contains moisture the filtering device must be heated to about 110 deg. cent. by any suitable means, but preferably by surrounding the brass shell with an electrically heated sleeve as shown in Fig. 3, which represents the sampling pipe, Brady filter, moisture tubes and gas meter assembled ready for use. The nipple on the outlet end of the brass cylinder is threaded so that it can be removed and replaced by aluminum

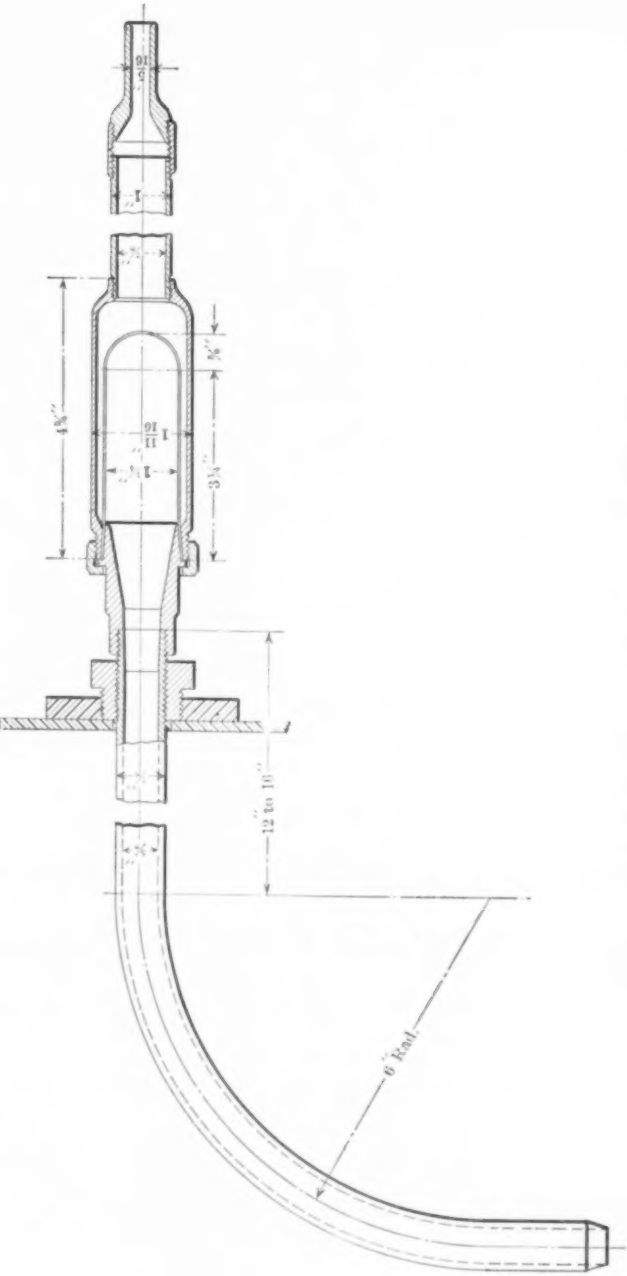


FIG. 2 SECTION OF BRADY DUST FILTER

tubes filled with a dehydrating agent such as calcium chlorid, in case it is desired to determine the moisture in the gas.

11 The Brady dust filter was chosen for use as standard instrument on account of its advantages over other methods. It is very simple, easily assembled and taken apart, perfectly tight, and it fulfills the requirement that the gas issuing from the sample pipe should pass over very little surface before being filtered. The principal advantage, however, is the use of a strong cylindrical filter which resists the gas pressure much better than the thin sheet of filter paper used in other instruments. The Soxhlet shell maintains a porous condition even though much fine dust has been deposited owing to the formation of concentric layers of dust and their subsequent cracking by the action of gravity and slight jarring. Gas samples of twice the size permissible in other instruments can be passed through the Brady filter. Fig. 4 shows five Brady filter shells after use for determination of dust at the main water seal, before and after the wet scrubbers, after the Theisen washers, and after the power station gas holder. The feature of keeping the filter porous for large samples is noticeable, particularly in the shell on the extreme left.

12 Whenever determinations of dust and moisture, or both, are desired, the instrument must be placed as close as possible to the pipe where the sample is to be taken. The sampling pipe used, shown in Fig. 3, consists of a $\frac{1}{2}$-in. brass pipe curved with a radius of not less than 6 in. and smoothly polished on the inside. It is inserted in the gas flue, if possible on a horizontal diameter at least 15 ft. away from any bend or obstruction, to a distance of one-fourth to one-third of the diameter. The inlet opening is reduced to a sharp edge, so that there is as little local disturbance at that point as possible. The question of the proper form of sampling pipe was decided in favor of the curved pipe, against the straight pipe with standard 4-in. insertion in the gas main. Experiments were made at various times to determine the amount of flue dust, by simultaneously using both forms of sampling pipe inserted at practically the same place in the gas flues.

13 A comparison of the results of these tests shows plainly the difference in the effect of straight and curved sampling pipes on the size of the gas sample, which generally speaking is larger with the curved pipe. This advantage is, however, of secondary importance, compared with the material increase in the dust contents recorded by sampling the gas with curved pipes. This increase is particularly noticeable in testing dry cleaned gas, and shows that the heavier particles of dust cannot easily be induced to change their direction of travel to enter the straight sample pipe at right angles, but pass by the opening, which is parallel with the gas stream. The difference in results averages nearly 100 per cent in favor of the curved sampling pipe.

14 In the clean gas tests this difference is considerably less marked and the average size of the gas sample obtained is even smaller, while the amount of dust recorded when using the curved sampling pipe is only $8\frac{1}{2}$ per cent larger, because the dust remaining in clean gas is of much finer quality and of less specific gravity, so that the particles will much more easily change their direction of travel. No difference at all, however, can be observed when dealing with fine gas. The averages of two series of tests coincide exactly, and only slight deviations are noticeable in the individual readings. The average size of the gas sample taken with the curved pipe is 14 per cent larger then the sample

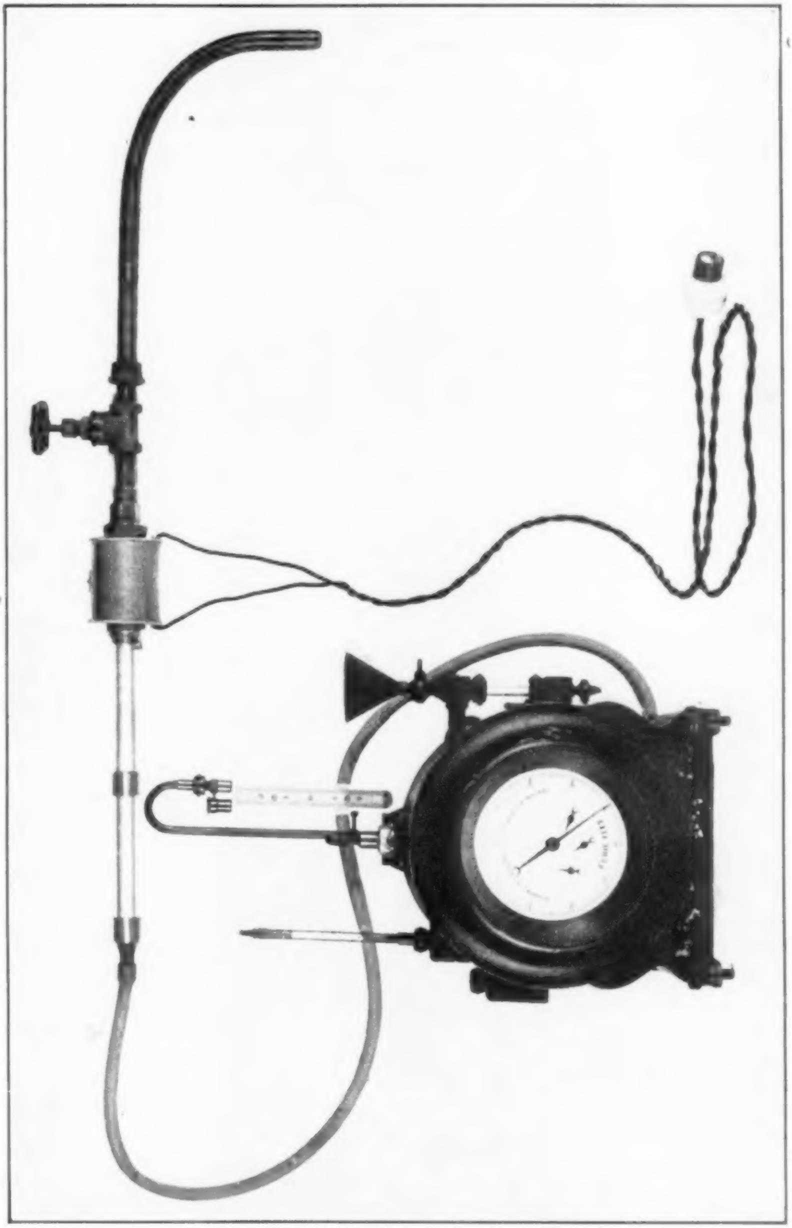


FIG. 3 BRADY FILTER, SAMPLING PIPE AND GAS METER

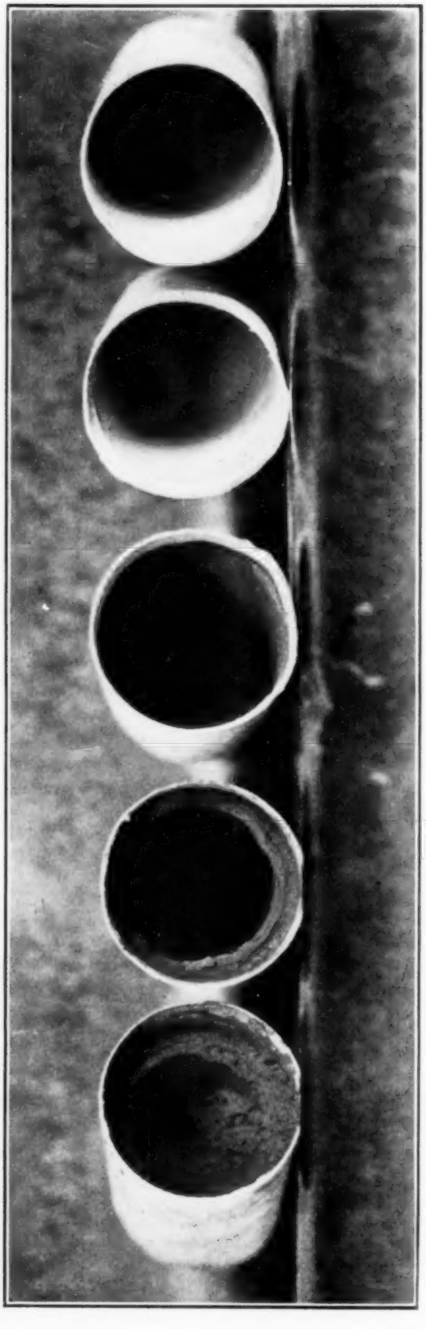


Fig. 4 Brady Filter Shells After Use

obtained with the straight sampling pipe. This experience is valuable as it shows that dust determinations in fine gas are very much less influenced by variations in the method of sampling, which further suggests that results will not be materially affected by variations in the instruments and methods used for dust determinations in fine gas.

15 As the original gas pressure is usually—in fine gas always— sufficient to cause the gas to flow through the filter, an aspiration of the sample is unnecessary. The determination is made as follows: The Soxhlet shell is first dried and weighed in a glass weighing bottle, then inserted into the brass shell of the apparatus, and the parts tightly connected. A meter reading is taken after the instrument is connected to the sample pipe and the gas is turned on. The dust contained in the gas sample is deposited in the Soxhlet shell, while the moisture is driven over into the aluminum tubes containing anhydrous calcium chlorid, and connected in series. They are capped and weighed before the experiment and their increase in weight represents the moisture in the volume of gas passed through the apparatus, while the increase in weight of the shell after drying gives the amount of dust carried in the gas sample. For dust determinations from 30 to over 200 cu.ft. of gas are passed, depending on conditions of pressure and locality, while for moisture determinations only from one to five cu. ft. are used. Usually one or several moisture tests may be made while one dust test is being run, simply turning off the gas for a moment when the aluminum tubes are inserted and again when they are withdrawn. Readings of the meter and of the gas conditions such as temperature and pressure must of course be made at the beginning and end of each test. All results of dust and moisture tests are calculated to grains per cubic feet of standard gas.

16 The application of two Brady filters permits continuous determination, a feature of great importance in gas power plants since it permits uninterrupted surveillance of the gas cleaning plant and of its efficiency. Continuous dust determinations are being made every day except Sunday, by sampling dry cleaned gas, clean gas and engine gas. A Brady filter is started at each place at 8.30 a.m., and the gas is allowed to pass until 4.30 p.m., when the Soxhlet shell is removed and a new one inserted, which is in continuous use from 5.00 p.m. until 8.00 a.m. the following morning. The average size of sample for day and night runs respectively, is from 60 to 90 cu.ft. of dry cleaned gas, 80 to 160 cu.ft. of clean gas, and 120 to 200 cu.ft. of engine gas. It is evident that such large accumulative samples must very nearly represent a true average of the amount of dust contained in the gas. Occasional dust and moisture determinations usually practiced in the majority of plants are of comparatively little value, as they do not give the true average conditions of the gas. Comparisons of results obtained at different gas-cleaning plants cannot and should not be made and credited, unless all instruments, methods, size of samples, duration of tests, etc., are identically the same. Standardization of the method of determining dust and moisture in industrial gases would benefit the gas engine industry at large, and the method used at this plant, which has been thoroughly and continuously tried under all conditions, is worthy of consideration as a basis for standardization.

APPENDIX NO. 4

RESULTS IN DETAIL OF OPERATION OF GAS-CLEANING PLANT

TABLE 1 HEAT LOSS OF GAS BY RADIATION

MONTHLY AVERAGES

1908	July	Aug.	Sept.	Oct.	Nov.	Dec.	Average
Gas cleaned, cu. ft. per min.	14,020	12,850	16,950	18,070	17,690	13,090	15,420
Temperature of gas at main water seal, deg. fahr....	426	410	303	312	299	329	346
Temperature of gas to scrubber No. 1, deg. fahr.....	210	196	202	168	150	133	177
Difference:							
Loss by radiation, deg. fahr....................	216	214	101	144	149	196	169
Reduction in per cent...	50.7	52.2	33.3	46.1	49.8	59.5	48.8

TABLE 2 WET SCRUBBER EFFICIENCY

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.-June
Flue dust in dry cleaned gas, gr. per cu. ft.....	0.4772	0.4787	1.2951	1.0335	1.1172	1.0804	0.9137
Flue dust in clean gas, gr. per cu. ft	0.0766	0.1224	0.2238	0.2178	0.2146	0.2389	0.1825
Difference........................	0.4006	0.3563	1.0713	0.8157	0.9026	0.8415	0.7312
Per cent removed by wet scrubbers	84.0	74.5	82.8	79.0	80.8	77.8	79.7

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Flue dust in dry cleaned gas, gr. per cu. ft.........................	0.7990	1.6124	4.0940	2.8794	2.4004	1.1188	2.1506	1.5330
Flue dust in clean gas, gr. per cu. ft...............................	0.1316	0.3669	0.8257	0.9882	0.2539	0.1786	0.4541	0.3183
Difference........................	0.6674	1.2455	3.2683	1.8912	2.1465	0.9402	1.6965	1.2147
Per cent removed by wet scrubbers..........................	83.5	77.4	79.9	65.7	89.6	84.2	78.8	79.3

TABLE 3 GAS AND WATER TEMPERATURES

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.-June
Temperature of gas to wet scrubber No. 1, deg. fahr	123.20	126.00	161.00	152.00	163.20	172.00	149.60
Temperature of gas to Theisen washers, deg. fahr	41.40	44.00	49.60	55.50	63.80	74.20	54.70
Temperature of gas to gas holder, deg. fahr	40.10	43.00	44.10	52.50	64.20	69.70	52.30
Temperature of water supply, deg. fahr	43.80	45.00	44.20	52.30	62.80	71.70	53.30
Waste, wet scrubber No. 1, deg. fahr	59.20	63.00	62.50	67.20	77.00	84.80	68.90
Waste, wet scrubber No. 2, deg. fahr	43.30	44.00	46.80	52.70	63.20	72.90	53.80
Temperature of air, deg. fahr	27.90	31.00	37.10	46.40	56.90	65.10	44.10

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Temperature of gas to wet scrubber No. 1, deg. fahr	169.20	173.60	150.60	159.50	119.30	86.70	143.30	145.70
Temperature of gas to Theisen washers, deg. fahr	76.50	79.10	72.70	61.80	61.70	52.80	67.40	61.10
Temperature of gas to gas holder, deg. fahr	79.10	80.90	72.30	61.00	61.10	42.00	54.40	56.60
Temperature of water supply, deg. fahr	74.70	78.60	72.80	63.70	64.50	46.20	66.70	60.00
Waste, wet scrubber, No. 1, deg. fahr	89.00	96.70	95.50	94.70	92.50	88.30	92.80	80.80
Waste, wet scrubber, No. 2, deg. fahr	76.70	81.50	77.10	64.90	65.40	52.70	67.70	61.70
Temperature of air, deg. fahr	83.40	85.20	74.60	60.80	58.80	31.20	65.60	54.90

TABLE 4 EFFICIENCY OF SECONDARY WASHING PLANT

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.-June
Flue dust in clean gas, gr. per cu. ft	0.0766	0.1224	0.2238	0.2178	0.2146	0.2389	0.1825
Flue dust in fine gas, gr. per cu. ft.	0.0036	0.0057	0.0044	0.0059	0.0067	0.0067	0.0055
Difference	0.0730	0.1167	0.2194	0.2119	0.2079	0.2322	0.1770
Per cent removed by refining	95.4	95.4	98.0	97.4	96.6	97.3	97.0

TABLE 4—CONTINUED.

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Flue dust in clean gas, gr. per cu. ft.	0.1316	0.3669	0.8257	0.9882	0.2539	0.1786	0.4541	0.3183
Flue dust in fine gas, gr. per cu. ft.	0.0057	0.0058	0.0080	0.0074	0.0069	0.0093	0.0061	0.0058
Difference	0.1259	0.3611	0.8177	0.9808	0.2470	0.1693	0.4480	0.3125
Per cent removed by refining	95.6	98.5	99.0	99.1	97.5	95.0	98.7	98.1

TABLE 5 DETERMINATION OF FLUE DUST AT DIFFERENT POINTS OF SECONDARY CLEANING PLANT

GRAINS PER CUBIC FOOT

1910	Turn	After wet scrubbers	Before Theisen washers	After Theisen washers	After gas holder
March 21	night	0.1496	0.1091	0.0024	0.0024
March 22	day	0.1681	0.1468	0.0066	0.0051
March 22	night	0.1647	0.1292	0.0045	0.0042
March 23	day	0.1773	0.1563	0.0091	0.0094
March 23	night	0.1185	0.1145	0.0062	0.0065
March 24	day	0.1566	0.1489	0.0058	0.0058
March 24	night	0.1456	0.1371	0.0066	0.0064
March 25	day	0.1568	0.0983	0.0067	0.0061
March 25	night	0.1474	0.1258	0.0064	0.0057
March 26	day	0.1425	0.1078	0.0037	0.0029
Average		0.1527	0.1274	0.0058	0.00545

Average amount removed
- By clean gas main .... 0.0253 grains per cu. ft.
- By Theisen washers .... 0.1216 grains per cu. ft.
- By fine gas main and gas holder .... 0.00035 grains per cu. ft.

Average absolute efficiency of
- Clean gas main .... 16.56 per cent.
- Theisen washers .... 95.45 per cent.
- Fine gas main and holder .... 6.03 per cent.

The average total efficiency of the secondary cleaning plant was 96.43 per cent, in which the three above factors participated as follows:
- Clean gas main .... 16.56 per cent.
- Theisen washers .... 79.64 per cent.
- Fine gas main and gas holder .... 0.23 per cent.

Total .... 96.43 per cent.

TABLE 6 DUST IN COMBUSTION AIR

Date	Wind	Day Turn: Temp of Air Deg. Fahr.	Day Turn: Barometer	Day Turn: No. cu. ft. sample	Day Turn: Dirt grams per cu.	Weather	Night Turn: Temp. of Air	Night Turn: Barometer	Night Turn: No. cu ft. sample	Night Turn: Dirt grams per cu. ft.
7-12-09	W.	78	29.21	74.13	0.0052	Part Cloudy	70	29.31	81.88	0.0043
7-13-09	W.	76	29.19	95.57	0.0032	Part Cloudy	71	29.26	57.59	0.0037
7-14-09	S. W.	80	29.27	124.51	0.0013	Wind blowing hard	73	29.28	102.84	0.0052
7-15-09	W.	84	29.28	126.41	0.0048	Part cloudy	76	29.28	129.11	0.0036
7-16-09	N. W.	80	29.34	88.69	0.0004	Wind hard	86	29.40	95.46	0.0005
7-17-09	N. W.	84	29.39	145.82	0.0004	Clear	..	......	......	....
Average				109.19	0.00255				93.37	0.00346

TABLE 7 ANALYSES OF DUST DEPOSIT ON GAS AND AIR DAMPERS OF GAS ENGINE NO. 1

Sample February 1909	Gas Damper		Air Damper	
	1	2	3	4
Silica	19.60%	22.50%	32.40%	23.80%
Alumina	12.07	20.19	6.50	11.30
Iron	6.95	6.37	11.12	12.03
Manganese	2.52	2.62	1.04	1.15
Lime	32.74	23.00	5.84	5.37
Magnesia	3.43	3.38	0.92	0.95
Volatile	17.89	17.38	37.84	39.85

TABLE 8 ANALYSES OF FLUE DUST REMOVED FROM BLAST FURNACE GAS AT DIFFERENT STAGES OF GAS CLEANING

March 1908	SiO_2	Al_2O_3	Fe	CaO	MgO	Fix C.	Sul.	Phos.	Mang.	Vo.
From main water seal (deposit)	10.30	4.60	48.85	2.46	0.30	15.52	....	0.067	0.53	....
From collecting main after dry cleaning plant (deposit)	11.44	4.45	42.86	3.15	0.68	10.28	0.202	0.079	0.80	7.28
From No. 1, wet scrubber (sediment)	14.58	5.45	43.09	2.75	0.78	9.17	0.288	0.095	0.60	4.17
From No. 2, wet scrubber (sediment)	18.26	5.85	38.20	4.74	1.40	8.54	0.192	0.097	0.47	5.08
From Theisen washers (sediment)	22.93	7.94	26.06	7.60	1.61	11.47	0.314	0.119	1.08	6.73

TABLE 9 ANALYSIS OF FLUE DUST REMAINING IN BLAST FURNACE GAS AT DIFFERENT STAGES OF GAS CLEANING

MARCH 1909

Location of Brady filters	SiO_2	Al_2O_3	Fe_2O_3	CaO	MgO	Mn.	Vol.
At main water seal	12.19%	5.93%	52.39%	4.70%	0.97%	1.16%	22.32%
Gas entering wet scrubber No. 1	11.37	5.21	52.55	3.76	0.86	0.83	25.18
In clean gas main	21.14	11.53	28.35	9.56	1.94	2.46	24.31

TESTS ON WET SCRUBBERS

1 Tests were made on October 27, 1908, to determine the cooling and condensing effect of the wet scrubbers. The temperature of the water and gas entering and leaving the washers were taken with accurate thermometers.

TABLE 10 WET SCRUBBER TEST

Time	Temperatures							Quantities			
	Water				Gas			Gas	Water		
	Scrubbers				Scrubbers			Cu. ft. per min.	Gal. per min.		
	Inlet 1 and 2	Outlet 1	Outlet 2	Outlet 1 and 2	Inlet 1	Outlet 1	Outlet 2		No. 1	No. 2	Total
10.50	63.0	80.0	64.0	73.0	157.0	62.8	60.0	17,000	....	....	1,380
11.00	62.8	77.5	63.0	71.5	156.0	62.8	60.0	17,150	....	....	1,375
11.10	63.0	77.5	63.2	72.0	156.0	62.8	60.0	17,350	....	....	1,375
11.20	63.0	76.5	63.5	71.5	155.0	62.8	60.0	17,150	....	....	1,375
11.30	62.8	77.2	63.0	71.5	155.0	62.8	60.0	17,150	....	....	1,375
Average....	62.9	77.74	63.34	71.9	155.8	62.8	60.0	17,160	816	560	1,376
2.55	63.6	85.5	63.0	77.0	180.0	64.8	62.0	23,100	....	....	1,225
3.05	63.8	85.5	63.0	76.5	177.0	64.8	62.0	23,400	....	....	1,275
3.15	63.8	85.5	63.0	76.0	175.0	64.8	61.5	23,100	....	....	1,225
3.25	63.5	85.0	63.0	76.5	175.0	64.3	61.5	23,250	....	....	1,290
3.35	63.5	87.0	63.2	77.0	176.0	64.3	61.5	23,100	....	....	1,260
Average....	63.64	85.7	63.04	76.6	176.6	64.5	61.7	23,190	752	503	1,255

The total amount of water from both washers was measured by the weir, and the amounts passing through each washer were calculated from the final temperatures. The gas was measured by venturi meter. The temperature of the atmosphere was 45 deg. fahr. Two tests of 40 minutes each were made on the same day. The readings and averages are given in the table.

2 The total heat absorbed in the first scrubber during the first test was 100,912 B.t.u. per min., of which 31,205 B.t.u. is accounted for in the loss of sensible heat in the gas. On the second test the total heat absorbed was 137,874 B.t.u. per min., of which 50,848 B.t.u. is accounted for by the loss of sensible heat. From the following calculations the amount of vapor condensed per cu. ft. of gas at 64 deg. was found to be 27.4 grains in the first, and 25.2 graids in the second test, or an average of 26.3 grains.

First Test:

Washer 1, temperature of entering gas, 155.8 deg.; leaving gas, 62.8 deg.

Density of gas = 0.0815

Specific heat = 0.24

Cu.ft. of gas per min. = 17,160

Sensible heat lost by gas:

$17,160 \times 0.0815 \times 0.24 \times (155.8 - 62.8) = 31,205$ B.t.u. per min.

Heat absorbed by water:

$816 \times 8\frac{1}{3} \times (77.74 - 62.90) = 100,912$ B.t.u. per min.

100,912 B.t.u. − 31,205 B.t.u. = 69,707 B.t.u.

Average latent heat of vapor from 155.8 deg. to 62.8 = 1037 B.t.u.

$\frac{69,707}{17,160 \times 1037} = 0.00392$ lb. or 27.4 gr. of vapor condensed per cu. ft. of gas at 64 deg.

Second Test:

Sensible heat lost by gas:

$23,190 \times 0.0815 \times 0.24 \times 112.1 = 50,848$ B.t.u.

Heat absorbed by water:

$752 \times 8\frac{1}{3} \times 22.06 = 137,874$ B.t.u.

173,874 B.t.u. − 50,848 B.t.u. = 87,026 B.t.u.

Average latent heat from 176.6 to 64.5 deg. = 1029 B.t.u.

$\frac{87,026}{23,190 \times 1029.7} = 0.003605$ lb. or 25.2 gr. vapor condensed per cu. ft. gas at 64 deg.

3 Since the average temperature of the gas leaving the first scrubber was about 64 deg., the amount of moisture remaining in the gas was about 6.6 grains; this added to 26.3 grains condensed gives 32.9 total grains of moisture per cu. ft. in dry cleaned gas. This represents a dewpoint of about 117 deg. or about 31 per cent saturation at the average initial temperature of 166 deg. Later tests with wet and dry bulb thermometers in the gas mains showed dewpoints varying from 104 deg. to 114 deg. for an average gas temperature of 170 deg. The results of these tests indicate the reducing effect which the washing of the gas has on the moisture. While by these calculations the amount of moisture in the dry cleaned gas was found to be about 33 grains per cu. ft. in October 1908, moisture determinations with Brady filters made on July 14, 15 and 16, 1909, gave very similar results. It will be noted that the test figures fairly coincide with the calculated values.

1908, moisture determinations with Brady filters made on July 14, 15 and 16, 1909, gave very similar results. It will be noted that the test figures fairly coincide with the calculated values.

TABLE 11 MOISTURE TEST IN DRY CLEANED GAS

Date	CO_2	CO	H	CH_4	B. t. u	$\frac{CO}{CO_2}$
July 14 Gas analysis	14.8	25.5	3.0	0.1	91.8	1.72
Grains of moisture per cu. ft.						34.124
Temperature at meter, deg. fahr.						84
Barometer, inches of mercury						29.27
Temperature in gas main, deg. fahr.						192
Pressure in gas main, inches of water						9.5
July 15 Gas analysis	13.8	25.7	4.0	0.1	95.3	1.86
Grains of moisture per cu. ft.						41.7453
Temperature at meter, deg fahr						85
Barometer, inches of mercury						29.68
Temperature in gas main, deg. fahr.						156
Pressure in gas main, inches of water						6.5
July 16 Gas analysis	13.1	26.1	3.5	0.2	96.1	1.99
Grains of moisture per cu. ft.						38.421
Temperature at meter, deg. fahr.						83
Barometer, inches of mercury						29.36
Temperature in gas main, deg. fahr.						190
Pressure in gas main, inches of water						10
Average moisture, gr. per cu. ft.						38.1

APPENDIX NO. 5

METHOD USED FOR MEASURING AND RECORDING GAS CONSUMPTION

The following description of the method used for measuring and recording the gas consumption was contributed by C. J. Bacon, Mem. Am. Soc. M. E.

2 The amount of gas consumed by the blowing engines at the blast furnaces and the power engines in the electric station, is measured by venturi meters one in the 54-in. main to the blowing engines and another in the 60-in. main to the power engines. The 60-in. meter was installed first and tested by volumetric measurements as hereinafter described. The 54-in. meter was subsequently constructed with the same proportions, and as the only difference is in the size no tests have been thought necessary. These meters are of much the usual form, except that certain liberties were taken in the design to simplify the shop work; the throat section of each being a straight cylinder connected to the small ends of the upstream and downstream cones without rounding at the intersections; and there was a similar omission of curvature at the connection between the approach section of 5ft. pipe and the large end of the upstream cone. Although it was realized that these departures from theoretically perfect design were likely to introduce more or less error due to eddy currents, nevertheless in view of the facility with which the accuracy could be determined by means of the gas holder, the somewhat irregular construction was allowed to stand. The absence of test data on meters of this size made tests advisable regardless of how nearly perfect the shape and construction might be.

3 By referring to Fig. 23*b* of the paper, it will be seen that the 60-in. meter has an over-all length of 53 ft. 1 in., and consists of an up-stream cone 11 ft. 6 in. long and having openings 60 in. and 20 in. in diameter, a straight cylindrical throat section of cast iron 20 in. in diameter by 15 in. long, and a down-stream cone 39 ft. long, likewise with openings 60 in. and 20 in. in diameter. The up-stream cones are made of plate, with butt-joints and countersunk rivets inside to reduce friction. A cylindrical casting 16 in. long by 60 in. in diameter and containing an annular pressure chamber, is inserted between the straight-approach pipe and the upstream cone. A similar pressure chamber surrounds the throat. Twelve 3/16-in. holes communicate to each of the chambers the pressures existing within the meter at those points. The characteristic equation for flow of gas in venturi tubes[1] was used in the calibration of this meter.

4 A number of carefully conducted tests have been made at various times to determine the meter coefficient, utilizing the 100,000 cu. ft. gas holder as a means of volumetric measurement. This holder is located about 260 ft. from

[1]See The Flow of Fluids in a Venturi Tube, by E. P. Coleman, Transactions, vol. 28, 1907 p. 483, for the derivation of this equation.

the meter, and is provided with a combination of water-sealed valves such that the flow of gas to and from the holder may be controlled at will. The horizontal area of the holder was accurately determined by measurement of diameters, and a vertical scale of feet and tenths was marked on the outside to permit of determination of the rate of rise of the holder. Observations were taken of

A gas pressure in the upstream chamber.
B difference in pressure between upstream and throat chambers.
C temperature of gas at meter and holder.
D analysis of gas including water vapor contents.
E barometric pressure.
F gas pressure at inlet to holder.

From these data and the dimensions of the meter, values may be assigned in the above-mentioned equation of flow. It is worthy of especial note that the ratio of specific heats for the mixture of gas and aqueous vapor is in this case 1.38, the use of it in the formulæ, however, does not result in an appreciably lesser flow than the use of 1.408, the commonly accepted value for air. Without burdening this paper with the actual data and computations, the net average results of 17 separate holder tests at various rates of flow shows a meter coefficient of 0.91, which is taken to mean that the actual flow is 91 per cent of the theoretical flow.

5 This determination of meter coefficient was made more as a matter of scientific interest than as a necessity, since working curves showing the relation between the difference in upstream and throat pressure and volume of gas at prevailing temperatures and pressures, could have been constructed from test data alone. The meter coefficient, however, being available, it was made use of in connection with the theoretical formula in preparing the curves in Fig. 35 of the paper, from which the volume of gas, reduced to standard conditions of 62 deg. fahr. and 29.92 in. barometer, may be determined for prevailing temperatures and meter readings. For these curves the absolute pressure of gas in the main is taken as 29.5 in. mercury, which is the sum of the average upstream pressure and the average barometer at this locality.

6 Daily records of flow of gas are obtained by means of a Bristol differential pressure recorder, located in the Theisen washer building about 250 ft. away from the meter and connected to upstream and throat chambers by two lines of pipe. Comparison of readings at the meter with the recorder shows no error due to the long connecting pipes. The curves of Fig. 2 are used in conjunction with these daily meter charts to obtain the average rate of flow for each day, which with the calorific value, the kilowatt output of the generators and the generator efficiency, gives the data required for computing the daily average thermal efficiency at the engine shaft. The chart shown in Fig. 36 of the paper represents a flow of 22.044 cu. ft. per min. Other observations and computations for that day were as follows:

B.t.u. per cu. ft. at 62 deg. fahr.	87.1
Average load, kilowatts	7306
B.t.u. per kw-hr	15761
B.t.u. per b.h.p-hr. at 96 % generator efficiency	11311
Thermal efficiency at shaft	22.5

The monthly averages of these daily data, and the results for the year 1909 are shown in Fig. 38 of the paper.

7 Questions are often raised regarding the amount of dust deposited in meters for blast furnace gas. The 60-in. meter has been examined at six-month intervals. The first inspection showed a slight accumulation of moist dust at and near the throat, but not in sufficient quantity to affect the results appreciably; at the second examination no dirt was found; at the third a considerable coating of dirt was found and was cleaned out, unfortunately without accurate measurement of the average thickness. As a means of determining the effect of the reducing diameter on the flow of gas, a comparison was made of the average thermal efficiency for a week preceding and a week following the cleaning as follows:

Week preceding cleaning	16.9 per cent
Week following cleaning	19.0 per cent
Reduction of flow	11 per cent

As far as known no change occurred at the engines to affect the efficiency; consequently, since the flow through the meter varies directly as the area, the reduced diameter due to dust was $\left(\frac{16.9}{19.0}\right)^{\frac{1}{2}}$ x 20 in. = 18.8 in. Therefore the average thickness of the coating was approximately 0.6 in.

8 The cause of this unusual deposit is ascribed to one of the furnaces making special irons, ferrosilicon and spiegel, during the latter part of August and the entire months of September and October 1909, or about 68 days during which the amount of dust found in the raw gas was excessive, as explained in par.73 of the paper. On this basis it is assumed that the deposit began late in August and continued at uniform rate through October, when the error amounted to a maximum of 11%. Suitable corrections were made on the monthly averages shown in the tables and charts. To prevent a repetition of the accumulations of dust in the venturi meter a system of spray nozzles (shown in Fig. 23*b* of the paper) was installed, for flushing the meter throat thoroughly with high-pressure water.

A COMPARISON OF LATHE HEADSTOCK CHARACTERISTICS

By Prof. Walter Rautenstrauch, New York

Member of the Society

The discovery of the properties of high-speed steels, and the large amount of experimental data available on the performance of these steels on various classes of materials, have urged the designer to attempt to incorporate in machine tools such characteristics as will adapt the machines to the most efficient use of the new steels. There exist at present many machines which are intended to meet the new standard of performance, and it will be interesting to examine the results of the attempts which have been made to meet the new conditions and to note the direction in which they have tended. There are many bases on which machine tools may be compared, and no single machine will ever prove best from all points of view; as the limits of this paper prevent the discussion of all these points, one of several possible standards will be adopted as a basis for comparison, and the results will be interesting though not conclusive.

2 Since the new steels will take heavier cuts than is possible with the carbon steels, and still retain their durability, a standard of comparison will be established on the basis of those characteristics of speed and torque in a lathe headstock which permit the most economic removal of shavings from a given class of material, viz., soft and medium steels. A comparison of the speeds and torques actually obtainable in any machine with the standard characteristics will serve as a means for judging the efficiency of the headstock in this particular. In this connection the method devised by Dr. J. T. Nicolson, of Manchester, is employed, the foundations of which are as follows:

3 Since the volume of metal removed by a lathe tool in a given time is a product of the area of cut and the speed of cutting, the weight removed in one minute will be equal to the area of cut in square inches times the speed of cutting in inches per minute times the weight of

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the metal per cubic inch. The force on the tool has been determined experimentally[1] to be approximately proportional to the area of the cut, the torque required to take any size cut is equal to the force on the tool times the radius of the work, and the speed at which the cut can be taken on any diameter of work depends on the spindle speed which can be obtained. These facts, together with the relations which have been established between possible maximum cutting speed and area of cut on different materials,[2] show that in any machine a definite relation must exist between the spindle speeds and the accompanying torques obtainable, that the machine may be adaptable to efficient weight removal on all diameters of any material.

4 The results of the experiments made by the Manchester Association of Engineers and the Berlin Section of the Verein Deutscher Ingenieure, have been used by Dr. Nicolson to derive equations expressing the approximate relation between the area of cut and the maximum cutting speed. The duration of cut was not less than 20 minutes, without injury to the tool. The following result was obtained for the materials in question (medium and soft steel):

$$V = \frac{1}{a} + 15 \qquad [1]$$

where V = cutting speed in feet per minute.
a = area of cut in square inches.

This equation, therefore, serves to determine the cutting speed at which it is possible to operate on this material without injury to the tool, when taking a cut of a given size.[3]

5 To establish a basis for determining the spindle speeds and torques required to remove the maximum weight of shavings on all diameters of work, it is necessary to determine the average area of cut which a lathe of given size should be expected to take. This was accomplished by Dr. Nicolson through correspondence with lathe builders, and the conclusion reached[4] was that the following rule met with wide acceptance for the machining of mild steel forgings:

$$a = \frac{S^2}{25{,}600} \qquad [2]$$

where a = area of cut in square inches.
S = swing of lathe in inches.

[1] Transactions, vol. 25, p. 656.
[2] Report of Manchester Association of Engineers, October 24, 1903.
[3] The Engineer (London), April 7, 1905.
[4] The Engineer (London), April 28, 1905.

6 If the above relations are true, namely, that the maximum possible cutting speed for mild steel varies with the area of cut as expressed in Equation 1; that the average area of cut on this material which a lathe of any given swing should be expected to accommodate is as given in Equation 2; and that the force in the tool varies directly as the area of the cut (for mild steel the force on the tool is approximately 100 tons for each square inch of area cut): then the following basis may be established for the design of, say, an 18-in. lathe capable of removing the maximum weight of shavings in a given time on all diameters of work.

$$\text{standard area of cut on all diameters} = \frac{S^2}{25{,}600} = \frac{18^2}{25{,}600} = 0.0126 \text{ sq. in.}$$

$$\text{Force on tool} = 100 \times 2000 \times 0.0126 = 2520 \text{ lb.}$$

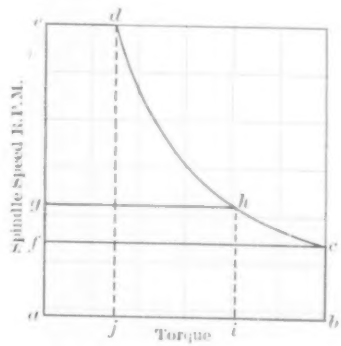

FIG. 1 IDEAL SPEED-TORQUE DIAGRAM

$$\text{torque on work of face plate diameter} = 2520 \times \frac{3}{4} = 1890 \text{ ft. lb.}$$

$$\text{maximum cutting speed at which cut may be taken} = \frac{1}{a} + 15$$

$$= \frac{1}{0.0126} + 15 = 94\tfrac{1}{2} \text{ ft. per min.}$$

revolution of spindle required for this cutting speed on work of face

$$\text{plate diameter} = \frac{94\frac{1}{2}}{\frac{3}{4} \times 2\pi} = 20 \text{ r.p.m.}$$

7 The maximum torque required of the lathe will on this basis be equal to 1890 ft. lb., while the minimum spindle speed necessary to give the maximum cutting speed on this area of cut at face plate diameter is 20 r.p.m. When the standard area of cut is taken on a smaller diameter the resulting torque will obviously be less. It will be

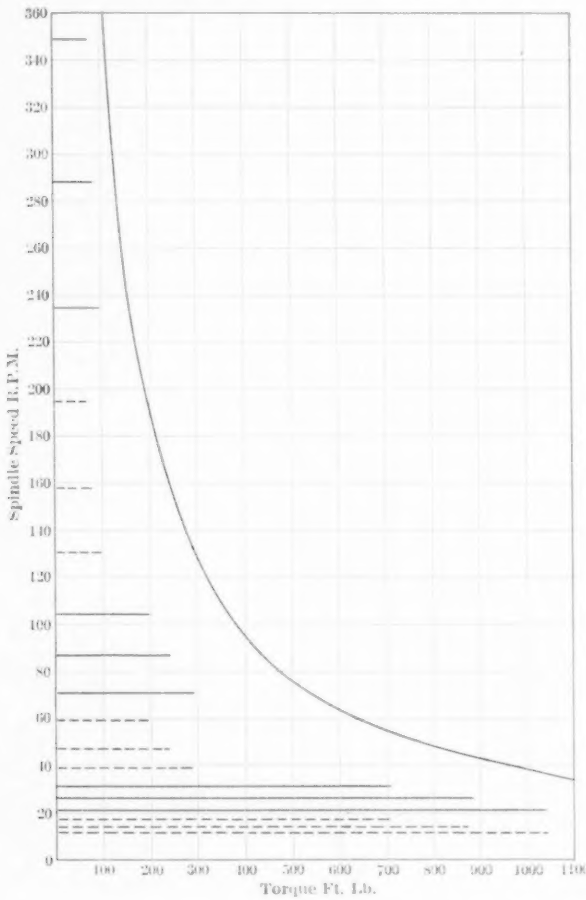

Fig. 2 Actual Speed-Torque Diagram

necessary, however, to increase the spindle speed of the machine, if that surface speed is to be maintained, which is desirable for the most efficient use of the tool and to maintain constancy of volume removed.

8 Since the spindle speed for these conditions will vary inversely as the diameter of the work, and the torque directly as the diameter

of work, it will be obvious that when the problem is to remove the maximum weight of shavings on all diameters of work, the product of speed and torque should be a constant. The highest spindle speed for which the lathe should be designed will depend on the smallest diameter of work which the lathe can economically handle, and the maximum cutting speed desirable on this diameter. On the basis that the least diameter is $\frac{S}{16}$, and that a cutting speed of 120 ft. per min. should be provided, the maximum spindle speed should be

$$Ng = \frac{12 \times 120}{\frac{S}{16}\pi} = \frac{7200}{S} \text{ (approx.)}$$

TABLE 1 (FIG. 2) 18-IN. LATHE

Countershaft Speeds 195 and 235 r.p.m.; Cones 13 in., 10¾ in., 8¾ in. Diameter; First Back-Gear Ratio 3.31 to 1; Second Back-Gear Ratio 10.95 to 1, Belt 3½ in.; Assumed Belt Pull 50 lb. per inch of Width

Spindle Speed r.p.m.	Torque ft. lb.	Spindle Speed r.p.m.	Torque ft. lb.
12.00	1040	71.00	314
14.40	865	87.30	261
17.80	700	105.40	212
21.46	1040	131.25	95
26.40	865	158.17	79
31.87	700	195.00	64
39.65	314	235.00	95
47.80	261	289.00	79
58.90	212	349.00	64

For the case of an 18-in. lathe this would result in a maximum spindle speed of 400 r.p.m.

9 In accordance with the above analysis, the ideal characteristic to which the design should tend is as shown in Fig. 1. The abscissæ represent torques, and the ordinates revolutions of the spindle. For the 18-in. lathe the dimensions of the diagram shown in Fig. 1 are

$ab = fc = 1890$ ft. lb.
$af = bc = 20$ r.p.m.
$ae = jd = 400$ r.p.m.
$aj = ed = 95$ ft. lb.

10 Since the product of speed and torque should be a constant, for the reasons previously explained, an equilateral hyperbola be-

tween the points d and c completes the construction of the ideal diagram. Accordingly, af is then the speed at which the spindle should run that the standard area of cut may be taken at its proper speed on work of face-plate diameter, and fc is the corresponding torque permitting this area of cut to be taken. Likewise, if the diameter of work is less than face-plate diameter, and since the torque varies directly as the diameter of work for a given area of cut, the torque for diameter of work equal to $S \frac{ai}{ab}$ and standard area of cut, is gh, while the spindle speed required to give the appropriate cutting speed is ag.

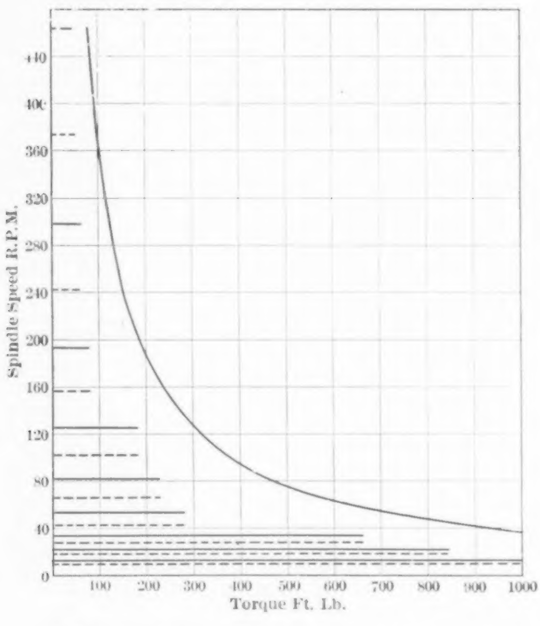

FIG. 3 ACTUAL SPEED-TORQUE DIAGRAM

11 It will be apparent that this diagram may be used in two ways; first, as a means for determining the proper relations which should exist between spindle speeds and torques when a lathe of any size is to be designed for conditions as defined above (to remove the maximum weight of shavings in a given time on all diameters of work of any given material); second, as a means for determining the extent to which the speeds and torques of a lathe already designed correspond to the standard established. In this latter connection it will also be

possible to determine whether or not any speeds, with their corresponding torques, might be omitted without hindering the weight-removing capacity of the headstock.

12 To illustrate the manner in which the diagram may be used as a standard for comparison, Figs. 2 and 3 are presented. In these figures are shown the speeds and torques obtainable in two lathes of recent manufacture, made by different firms. The data for the determination of the speeds and torques, which were obtained from the manufacturers' catalogues, are given in Tables 1 and 2.

13 On the basis that these lathes should be capable of operating on mild steel, with an area of cut on all diameters as determined in the above analysis and up to the maximum cutting speed which the

TABLE 2 (FIG. 3) 18-IN. LATHE

COUNTERSHAFT SPEEDS 196 TO 234 R.P.M.; CONES 12 IN., 9¾ IN., 7¾ IN. DIAMETER; FIRST BACK GEAR, RATIO 3.66 TO 1; SECOND BACK GEAR, RATIO 13.5 TO 1; BELT 3 IN.; ASSUMED BELT PULL 50 LB. PER INCH OF WIDTH

Spindle Speed r.p.m.	Torque ft.lb.	Spindle Speed r.p.m.	Torque ft.lb.
11.6	1012	82.0	226
14.4	1012	102.0	177
18.0	835	126.0	177
22.3	835	157.0	75
27.8	655	195.0	75
34.5	655	243.0	61.6
42.7	274	300.0	61.6
53.0	274	375.0	48.5
66.0	226	465.0	48.5

durability of the tool steel will permit, it will be noted that these designs are deficient; for example, if it were required to turn a piece of mild steel 9 in. in diameter, with a cut of 0.0126 sq. in. = $\frac{1}{8}$ in. x $\frac{3}{32}$ in. (approximately) the torque required would be

$$0.0126 \times 2{,}000{,}000 \times \frac{9}{2 \times 12} = 985 \text{ ft. lb.}$$

The lathe illustrated in Fig. 2 would have to take this on spindle speed 1 or 4. Speed 4 would give the highest cutting speed which would be

$$21.46 \times \frac{9 \times \pi}{12} = 50\tfrac{1}{2} \text{ ft. per min.}$$

But with the above area of cut a cutting speed of 94½ ft. per min. would be possible under ideal conditions, hence the minimum time in

which one pound of shavings could be removed under the actual circumstances is about twice what it would be if the required torque were available at the maximum cutting speed.

14 Any number of examples could thus be worked out to illustrate the limits which the dimensions of this headstock impose on either the area or speed of the cut which can be taken on any diameter of work. The question may be asked, to what extent do each of these speeds, with their corresponding torques, contribute to the weight-removing capacity of the lathe, when operating on this material?

15 Referring to Fig. 1, it will be noted that the area $ab \times bc$ is the product of the torque and spindle speed and is

$$f\ a\ r \times \frac{V}{2\pi r} = \frac{f}{2\pi}\ a\ v = K\ a\ V$$

where

f = force on tool in pounds per square inch.
a = area of cut in square inches.
V = cutting speed in feet per minute.
r = radius of work in feet.

But the area of the cut times the speed of cutting is a measure of the volume of metal removed in a given time and hence a measure of the weight removed in a given time. Any condition, therefore, fixing the limits to the area and speed of cut which can be taken on any diameter of work will limit the maximum rate at which metal can be removed.

16 Let us determine, therefore, to what extent the gap between the speeds, and the departure of the torque from that which has been established as desirable at the different speeds, will effect the weight-removing capacity of the lathe. Let Fig. 4 represent the ideal torque-speed diagram for any lathe, established on the above basis, and ab and ac two spindle speeds actually obtainable with torques bd and ec respectively. Then with a speed of spindle ab and torque bd, the standard area of cut may be taken on work of diameter $S\ \frac{bd}{ag}$, where S is the swing of the lathe.

17 Suppose it is only necessary to take a lighter cut $\left(a = \frac{bj}{bd}\right)$ on the same diameter of work, can it be more economically removed by taking the full cut $\left(a = \frac{bj}{bd}\right)$ with the spindle speed ab or, neglect-

ing the time for resetting the tool, to use a still lighter cut with the spindle speed *ac* and go over the work twice to bring it to finished size? Also, up to what limit of area of cut will it be more economical to use the speed *ab* than *ac*?

18 Now the whole area of cut may be taken at the lower speed *ab*, for which the rate of weight removed is represented by $(bj \times jq)$, or it may be taken at the higher speed *ac* by going over the work twice, first with a depth of cut and feed, giving an area of cut equal to $a_s \frac{bx}{bd}$, and again with remaining depth of cut and a feed giving the same area of cut $a_s \frac{bx}{bd}$, required to bring the piece down to size.

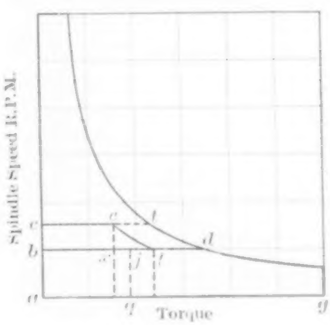

FIG. 4 IDEAL SPEED-TORQUE DIAGRAM

Neglecting for the present the time taken to reset the tool, it will be seen that the weight removed in the same time will be greater by going over the piece twice, each time with an area of cut equal to $a_s \frac{bx}{bd}$, and spindle speed *ac*, than by taking the cut of area $a_s \frac{bj}{bd}$ at the lower spindle speed *ab*.

19 To illustrate more specifically, suppose for example that *bd* represents the torque required to take the standard cut a_s on some given diameter of work, then *bj* would represent the torque required on the same diameter of work when the area of cut is not equal to the standard area but is equal to $a_s \frac{bj}{bd}$, since the ratio of the torques is equal to the ratios of the areas of cut on the same diameter of work.

If $a_s = 0.0126$ sq. in., $a_s \dfrac{bj}{bd} = 0.0084$ sq. in., $a_s \dfrac{bx}{bd} = 0.0063$ sq. in., and $ab = 30$ r.p.m., $ac = 50$ r.p.m., then the rate of weight removal when taking the area of cut $a_s \dfrac{bj}{bd}$ at the spindle speed ab is proportional to $0.0084 \times 30 = 0.252$, while if the area of cut $a_s \dfrac{bx}{bd}$ is taken at the spindle speed ac the rate of weight removal is proportional to $0.0063 \times 50 = 0.315$. The above condition will be true up to such areas of cut on the given diameter which, when multiplied by the lower spindle speed, will give a rate of weight removal greater than 0.315. This limit of area of cut may be conveniently determined by drawing an equilateral hyperbola through e and letting it cut bd at l.

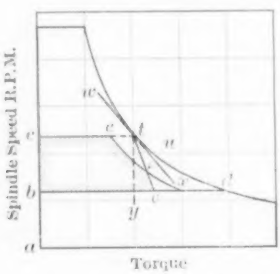

Fig. 5 Ideal Speed-Torque Diagram

Areas of cut from $a_s \dfrac{bl}{bd}$ to a_s can be more economically removed for the given diameter of work at the lower speeds, while areas of less than $a_s \dfrac{bl}{bd}$ can be more economically removed by taking the lighter areas of cut $a_s \dfrac{bx}{bd}$ at the higher spindle speed and going over the work twice to bring it to size. Therefore, the efficiency in weight removed of this range of speeds and torques, compared to the ideal case where all speeds and torques, define within the area $bctd$ are available, is represented by the ratio

$$\frac{\text{area } bcel}{\text{area } bctd}$$

In case the areas of cut from $a_s \frac{bx}{bd}$ to $a_s \frac{bl}{bd}$ can not be taken on the diameter in question at the higher spindle speed because the resulting surface speed is too great, the above statement is not true. A few of the designs examined have been checked in this manner and found to come within the limit just defined.

20 No allowance, however, has been made for the time required to run the carriage back and reset the tool. It will be seen that as the limiting area $a_s \frac{bl}{bd}$ is approached, the time saved on the use of the lower speed in place of the higher becomes less. Accounting for the time required to reset the tool for a second run, it will be noted that the limiting area is reached before $a_s \frac{bl}{bd}$. Just where the limit will be encountered it is impossible to determine except by empirical methods. Dr. Nicolson has ascertained that this limit may be approximately determined by the use of the following construction, irrespective of the type or design of the lathe.

21 Let Fig. 5 represent the conditions taken in Fig. 4. Construct a tangent *ux* to the hyperbola at *t* and drop the vertical *ty*. Bisect the angle between *ux* and *ty* by the line *tv*. The efficiency of this particular part of the headstock will be approximately represented by

$$\frac{\text{area } cevxb}{\text{area } ctdb}$$

Areas of cut equal to and greater than $a_s \frac{bv}{bd}$ can be more economically taken on this diameter of work at the lower speed *ab* because of the difference in time required to handle the machine for the two cuts required to bring the piece to size.

22 This constructionis to be considered as a rough approximation only, and represents the facts as well as the conditions in the case will permit. This method of comparing the efficiency of a lathe with a predetermined ideal performance on any given material is due to Dr. J. T. Nicolson and Mr. Dempster Smith, to whom all credit should be given. The above method is useful in determining the adaptability of a lathe to meet only one of the many kinds of service in which the lathe may be employed and is not a final means for either justifying or condemning a lathe for general purpose work.

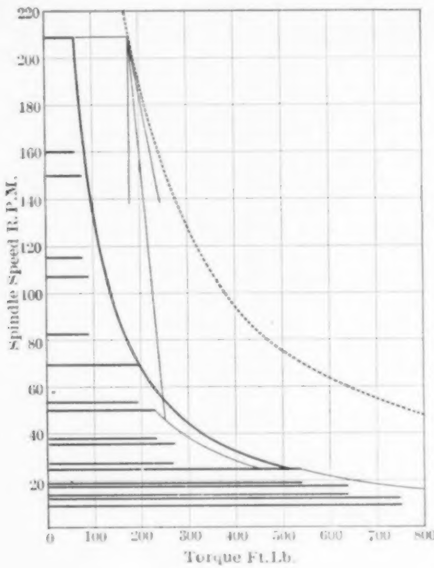

Fig. 6 16-In. Double Back-Geared Lathe

TABLE 3 (FIG. 6) 16-IN. DOUBLE BACK-GEARED LATHE

Cone Diameters $8\frac{3}{4}$ in., $9\frac{7}{8}$ in., $11\frac{1}{8}$ in.; Belt Width $3\frac{1}{4}$ in.; Countershaft Speeds 115 and 150 r.p.m. First Back-Gear Ratio 3 to 1; Second Back-Gear Ratio $8\frac{1}{4}$ to 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
9.87	748	50.00	231
12.85	748	53.50	193
13.80	642	69.60	193
18.00	642	82.50	90
19.20	537	107.00	90
25.00	537	115.00	77
27.50	270	150.00	77
35.73	270	160.00	65
38.33	231	209.00	65

The torques were computed on the basis of 50 lb. per inch of belt effective on the pulley surface. As a basis of the foregoing analysis, the lathe should be capable of the following:

Ng (greatest desirable spindle speed) = 450 r. p. m.
Ml (least desirable spindle speed) = $28\frac{3}{4}$ r. p. m.
Moximum desirable torque = 1366 ft. lb.

23 There is, however, one point of broad application which a speed torque diagram constructed according to the above basis will immediately bring out; that is, the uselessness of certain speeds, with their corresponding torques, possible in a given lathe on any class of work. As an illustration of how the relative merits of lathes of different make may be determined with reference to a common standard, 11 lathes selected from the catalogues of different builders have been used in the

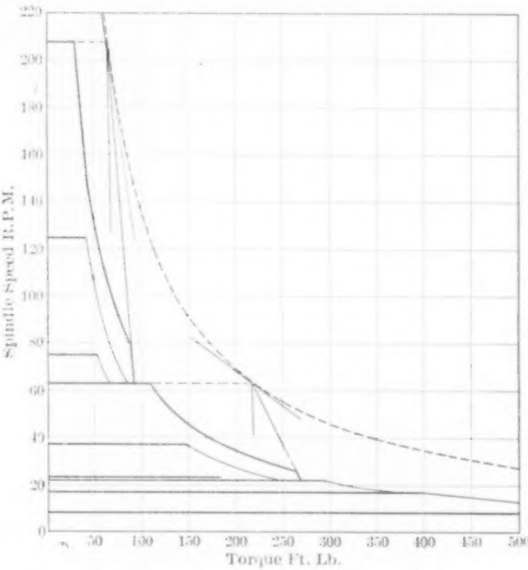

Fig. 7 16-in. Double Back-Geared Lathe

TABLE 4 (FIG. 7) 16-IN. DOUBLE BACK-GEARED LATHE

Cone Diameters 6 in., 8 in., 10 in.; Belt Width 2½ in.; Countershaft Speed 125 r.p.m : First Back-Gear Ratio 3½ to 1; Second Back-Gear Ratio 9½ to 1

Spindle Speeds r.p.m.	Torque ft.lb.
7 9	495
17 04	400
21 9	294
22 5	182
37 5	147
63 00	109
75 00	52
125 00	42
208 3	31

Ng = 450 r.p.m.; Nl = 28¾ r.p.m.; maximum torque = 1366 ft. lb.

construction of the following figures. In each case the data were obtained from the catalogues, or by correspondence with builders, and the possible speeds and torques determined. The data and results thus obtained are shown in Tables 3 to 13, the corresponding speed-torque diagrams being represented by Figs. 6 to 16.

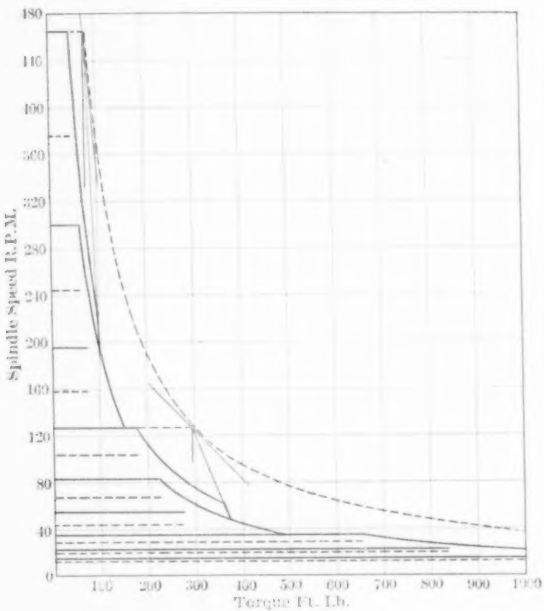

FIG. 8 18-IN. DOUBLE BACK-GEARED LATHE

TABLE 5 (FIG. 8) 18-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS 7¾ IN., 9⅞ IN., 12 IN.; BELT WIDTH 3 IN.; COUNTERSHAFT SPEEDS 196 AND 243 R.P.M.; FIRST BACK-GEAR RATIO 3.66 TO 1; SECOND BACK-GEAR RATIO 13.5 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
11.6	1012	82.0	226
14.4	1012	102.0	177
18.0	835	126.0	177
22.3	835	157.0	75
27.8	655	195.0	75
34.5	655	243.0	61.6
42.7	274	300.0	61.6
53.0	274	375.0	48.5
66.0	226	465.0	48.5

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft. lb.

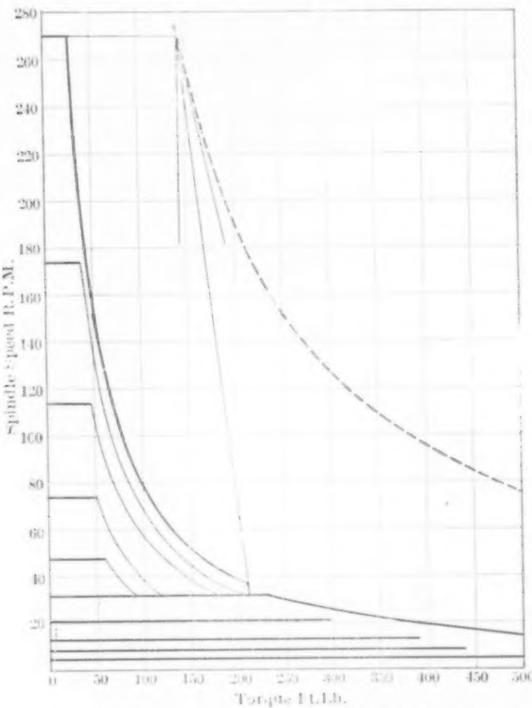

FIG. 9 18-IN. SINGLE BACK-GEARED LATHE

TABLE 6 (FIG. 9) 18-IN. SINGLE BACK-GEARED LATHE

CONE DIAMETERS $5\frac{1}{4}$ IN.; $6\frac{13}{16}$ IN., $8\frac{3}{8}$ IN., $9\frac{13}{16}$ IN., $11\frac{1}{4}$ IN.: BELT WIDTH $2\frac{1}{2}$ IN.; COUNTERSHAFT SPEED 125 R.P.M.; BACK-GEAR RATIO 8.44 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
5.70	500
8.75	437
13.50	390
20.60	300
32.00	231
48.00	60
74.00	52
114.00	46
174.00	36
270.00	27

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft.lb.

24 Among the facts brought out by this method of comparison of the adaptability of different makes of lathes to the performance of a standard task, there are two which are particularly striking. It will be noted in the first place that a considerable difference of opinion exists among the several builders, the characteristics of whose lathes

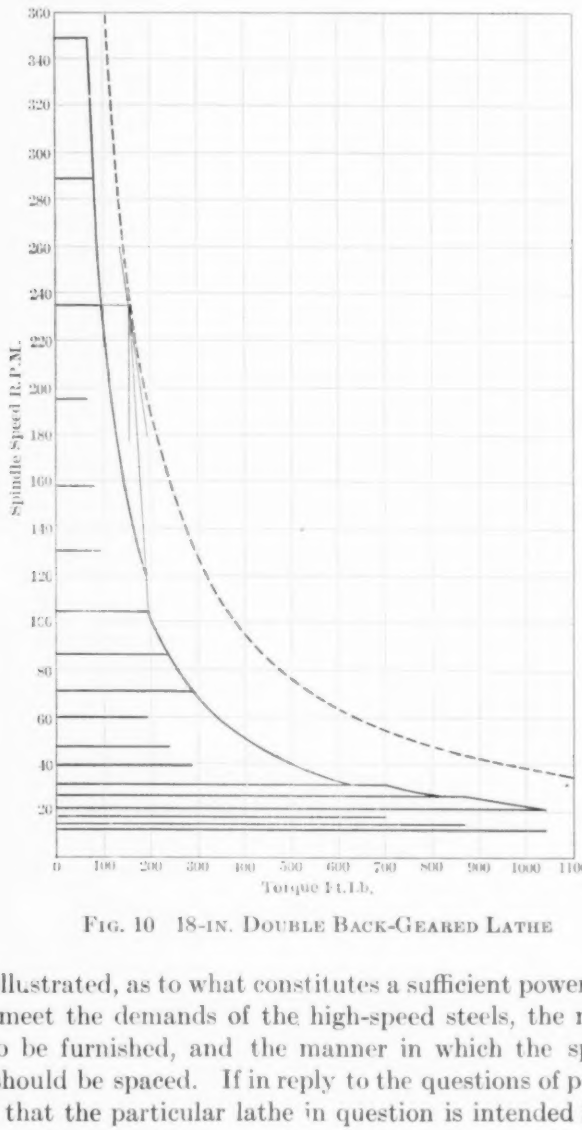

FIG. 10 18-IN. DOUBLE BACK-GEARED LATHE

are here illustrated, as to what constitutes a sufficient powering of the lathe to meet the demands of the high-speed steels, the number of speeds to be furnished, and the manner in which the speeds and torques should be spaced. If in reply to the questions of powering it is stated that the particular lathe in question is intended for taking

lighter cuts, which might be a proper basis for design under certain circumstances, it still remains to justify the manner in which the speeds and torques are spaced.

25 For example, take the case of the lathe represented in Fig. 6. For the single instance of having to turn a 9-in. piece of soft steel it

TABLE 7 (FIG. 10) 18-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS $8\frac{3}{4}$ IN., $10\frac{7}{8}$ IN., 13 IN.; BELT WIDTH $3\frac{1}{2}$ IN.; COUNTERSHAFT SPEEDS 195 AND 235 R.P.M. FIRST BACK-GEAR RATIO 3.31 TO 1; SECOND BACK-GEAR RATIO 10.95 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
12.0	1040	71.00	314
14.4	865	87.3	261
17.8	700	105.4	212
21.46	1040	131.25	95
26.4	865	158.17	79
31.87	700	195.00	64
39.65	314	235.00	95
47.8	261	289.00	79
58.9	212	349.00	64

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft. lb.

TABLE 8 (FIG. 11) 20-IN. ROUGHING LATHE

CONE DIAMETERS $11\frac{1}{2}$ IN. AND 13 IN.; 6-IN. DOUBLE BELT; COUNTERSHAFT SPEEDS 340 AND 365 R.P.M.; FIRST BACK-GEAR RATIO 3 TO 1; SECOND BACK-GEAR RATIO 6 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
50	1360	128	680
53	1210	137	605
64	1360	300	227
68	1210	323	202
100	680	384	227
107	605	412	202

Ng = 360 r.p.m.; Ml = 16 r.p.m.; maximum torque = 2666 ft. lb. Double belts are estimated as having 75 lb. per inch of width effective on pulley surface.

will be seen that the maximum area of cut that can be taken is limited to 0.01 sq. in., equivalent to a cut $\frac{1}{6}$ in. by $\frac{1}{16}$ in., and that the highest speed which the resulting torque of 750 ft. lb. will permit is 12.85 r.p.m., giving a cutting speed of 30 ft. per min. on this diameter. The cutting speed possible with soft steel on this area of cut is approximately 115 ft. per min. or if an area of cut of 0.0036 sq. in. is to be

taken on the same diameter, the highest spindle speed which the resulting torque of 270 ft. lb. will permit is 35.73 r.p.m., giving a cutting speed of 85 ft. per min. The cutting speed possible with this area of

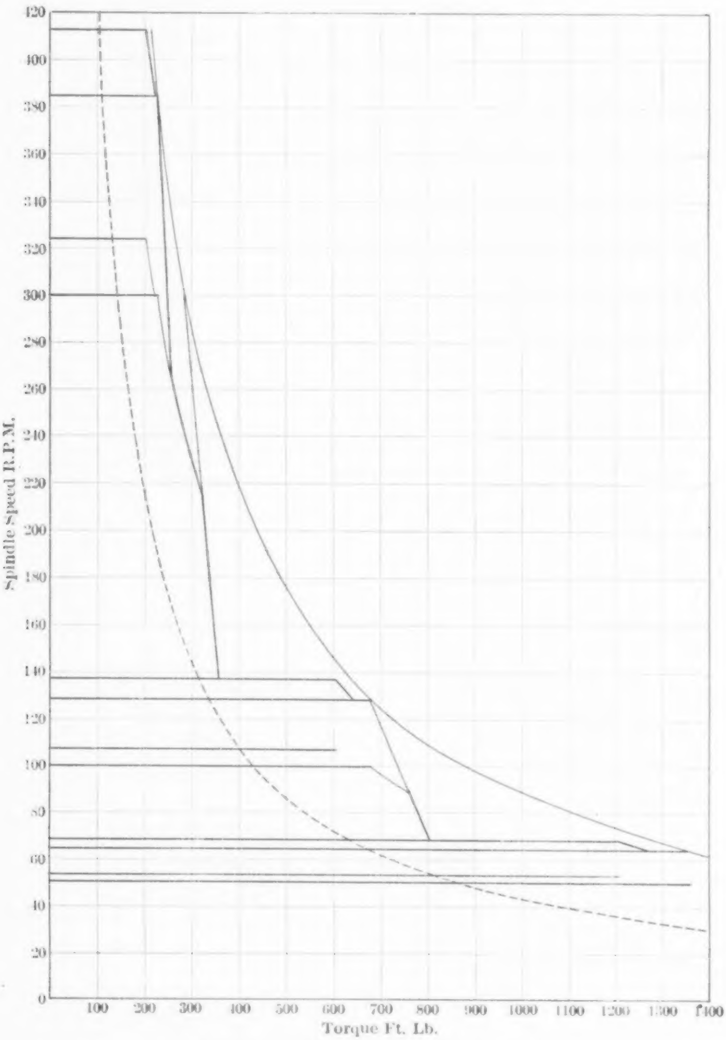

FIG. 11 20-IN. ROUGHING LATHE

cut is above 200 ft. per min. For this size of work, then, the lathe is inefficient, or for efficient operation is limited to forms of work in which the cutting speeds and area of cut determined are the highest

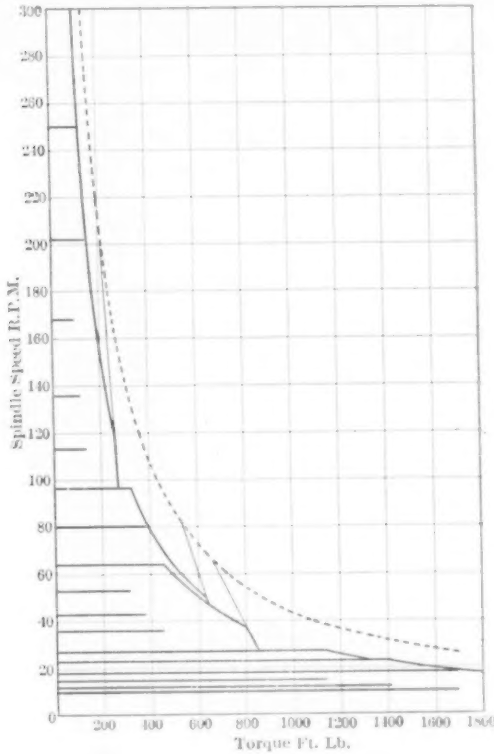

FIG. 12 21-IN. HEAVY-DUTY LATHE

TABLE 9 (FIG. 12) 21-IN. HEAVY-DUTY LATHE

CONE DIAMETERS 10¾ IN., 13¾ IN., 16 IN.; BELT WIDTH 4½ IN.; TWO COUNTERSHAFT SPEEDS; FIRST BACK-GEAR RATIO 3.13 TO 1; SECOND BACK-GEAR RATIO 11.3 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
12	1,700	64	470
10	1,420	80	394
15	1,150	96	320
18	1,700	113	150
23	1,420	136	126
27	1,150	168	102
36	470	202	150
43	394	250	126
53	320	300	102

Ng = 342 r.p.m.; Nl = 14 r.p.m.; maximum torque = 3087 ft. lb.

possible. In like manner, the limits of performance on any other diameter of work imposed by the torque-speed characteristics of the lathe, may be determined.

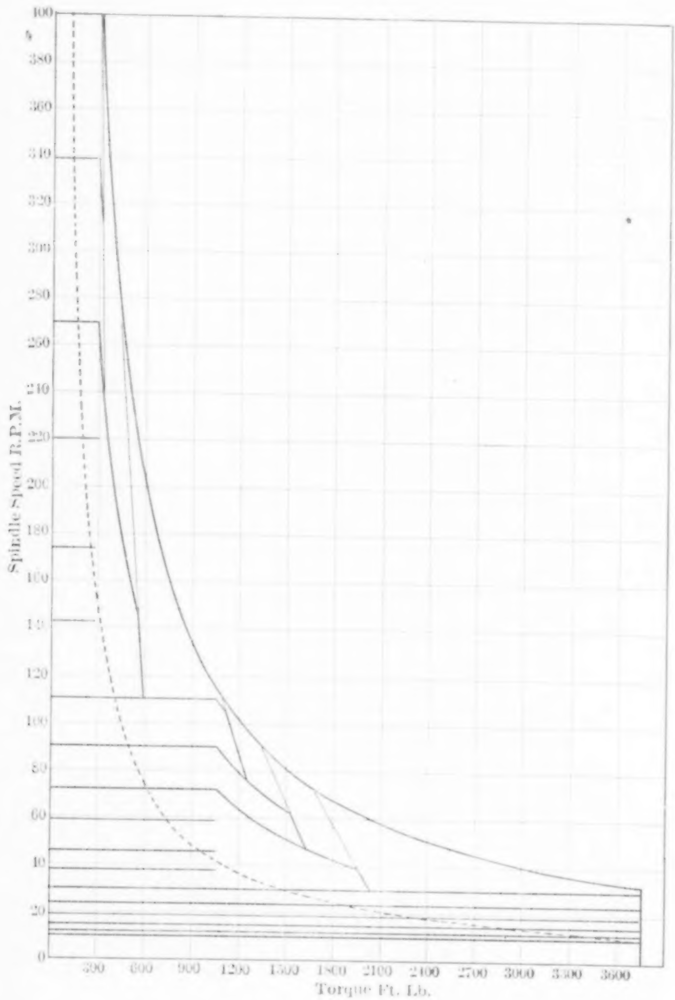

FIG. 13 21-IN. GEARED-HEAD LATHE

26 The extent to which the several speed-torque characteristics supplement one another is also very conveniently brought out in these diagrams. Again referring to Fig. 6, it will be noted that the con-

tribution of a number of the speed-torque combinations to the efficiency of the lathe for weight removal is brought into question, no matter what the standard of performance may be. If the foregoing analysis is rational, it indicates that the speeds 160, 150, 115, 107, 82, 73, 53, 50, 38, 35, 27, 19, 20, 18, 13.8, 12.85, and 9.87, with their accompanying torques are superfluous. Only upon a sufficient increase in the

TABLE 10 (FIG. 13) 24-IN. GEARED-HEAD LATHE

COUNTERSHAFT PULLEY 16 IN.; HEADSTOCK PULLEY $15\frac{13}{16}$ IN.; COUNTERSHAFT SPEEDS 205 AND 250 R.P.M.; $6\frac{1}{2}$-IN. DOUBLE BELT; FIRST BACK-GEAR RATIO 3.69 TO 1; SECOND BACK-GEAR RATIO 13 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
10	3760	72	1066
12	3760	90	1066
15	3760	110	1066
19	3760	143	289
24	3760	174	289
30	3760	220	289
38	1066	270	289
46	1066	339	289
59	1066	414	289

Ng = 300 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

TABLE 11 (FIG. 14) 24-IN. GEARED-HEAD LATHE

COUNTERSHAFT PULLEY 16 IN.; HEADSTOCK PULLEY 16 IN.; BELT WIDTH 5 IN.; COUNTERSHAFT SPEED 400 R.P.M.; BACK-GEAR RATIO 5 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
21	3174
32	2083
40	1666
60	1111
107	623
160	417
200	334
300	223

Ng = 300 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

corresponding torque can each of these speeds add to the efficiency of the lathe. Considered on the basis of a dead investment alone, it will be seen that the equipment required to give the above speeds, which seem without justification, adds a useless burden to the product of this machine.

27 An examination of some of the following diagrams will reveal facts similar to those announced above. In those cases where two countershaft speeds are employed it will be noted that no increase in efficiency is had from this source. It is a pleasure, however, to note some exceptions, particularly in the case of Fig. 14. It will be observed that upon this basis of analysis there is a justification for each speed-torque characteristic. If any of the speeds were cut out, the efficiency

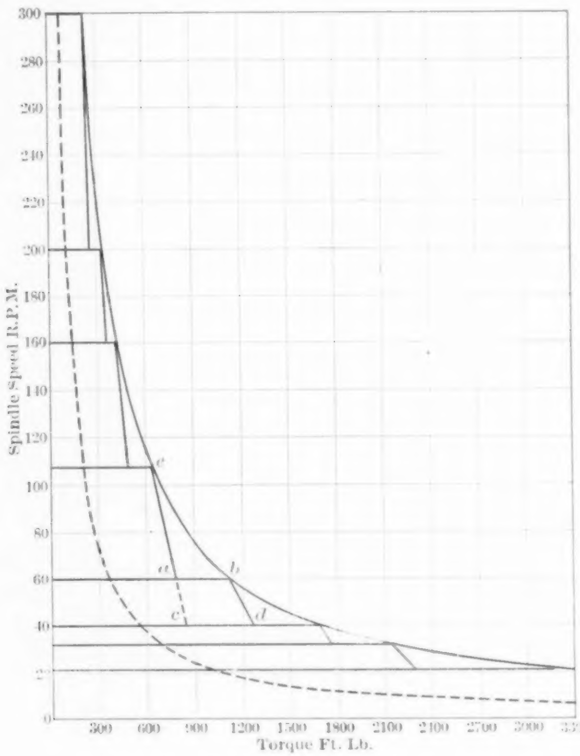

FIG. 14 24-IN. GEARED-HEAD LATHE

of the lathe would be reduced. With respect to the standard task used in this discussion, it will be noted that the removal of speed 60 from the headstock would reduce the efficiency by an amount proportional to the area *abcd*.

28 Another matter which appears in this connection is the relation of the efficiency to the number of speed-torque combinations, of which only eight are possible in this lathe. To what extent would the effi-

ciency be increased if eight additional speeds, with their accompanying torques, should be spaced halfway between the present combinations? The answer to this question would be obtained by the same method

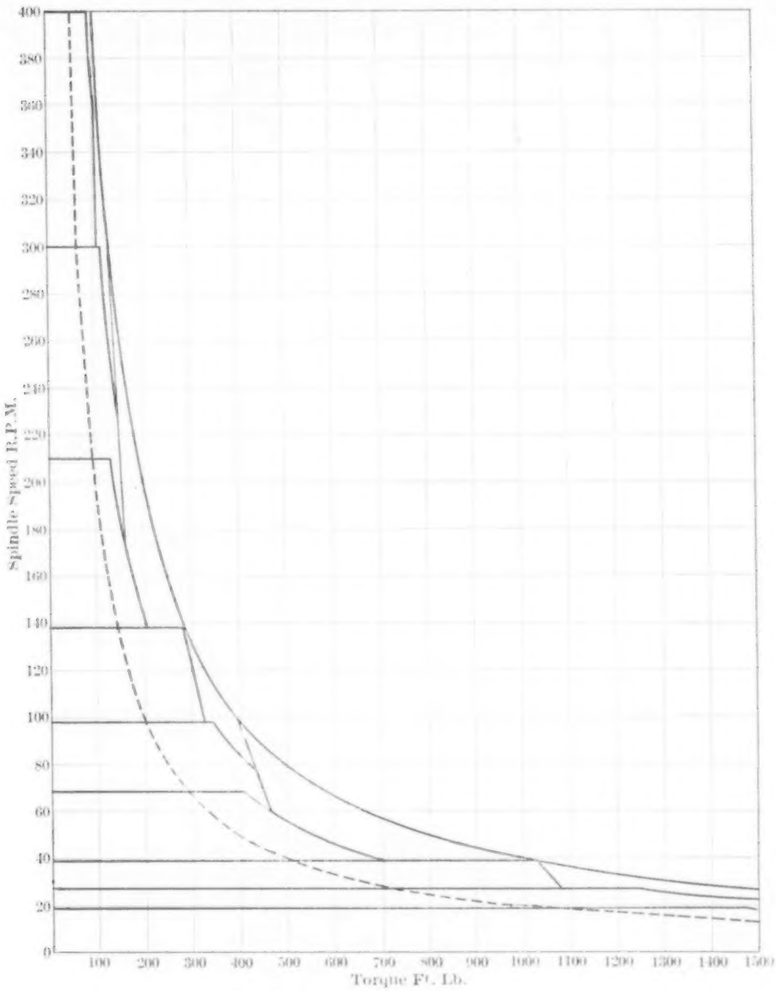

Fig. 15. 24-in. Double Back-Geared Lathe

by which it was determined that the omission of speed 60 in the previous problem would reduce the efficiency proportional to the area *abcd*.

29 An analysis of this sort will show two things: first, that increasing the number of speeds without regard to the torque does not necessarily increase its adaptability to economic performance; second, that the amount by which the efficiency can be increased does not increase in direct proportion to the additional amount of speed changes provided,

TABLE 12 (FIG. 15) 24-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS 10½ IN., 12¾ IN., 15 IN; BELT WIDTH 4½ IN.; COUNTERSHAFT SPEED = 300 R.P.M. FIRST BACK-GEAR RATIO 3.1 TO 1; SECOND BACK-GEAR RATIO 11.1 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
19	1476
27	1254
39	1032
68	410
98	350
139	288
210	133
300	113
429	93

Ng = 380 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

TABLE 13 (FIG. 16) 26-IN. "MASSIVE" LATHE

CONE DIAMETERS 7 IN., 9¾ IN., 12½ IN., 15¾ IN., 18 IN.; BELT WIDTH 4 IN.; COUNTERSHAFT SPEEDS 125 R.P.M.; BACK-GEAR RATIO 12 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
4.05	1800
6.65	1512
10.40	1250
16.3	975
26.8	700
48.6	150
80.0	126
125.0	104
196.0	81½
322.0	58¾

Ng = 277 r.p.m.; Nl = 8.15 r.p.m.; maximum torque = 5860 ft. lb.

even if the accompanying torques are properly determined. If 24 speed-torque combinations were properly spaced in the design represented in Fig. 14, the increase in efficiency over the eight already presented would not be twice as much as if 16 speed-torque combinations should be introduced in the same manner.

30 Closely associated with the matter of the increase in efficiency by the introduction of additional speed-torque changes is the problem of whether or not the increase is warranted by the increase in cost due to the additional equipment, and whether the management of the shop is such as to insure proper use of the additional equipment. The latter is in general the more vital question. In fact, the whole matter

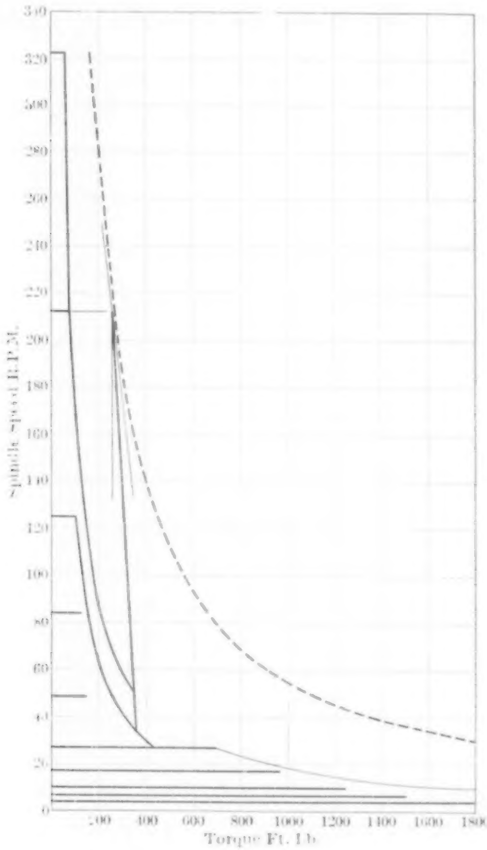

Fig. 16 26-in. "Massive" Lathe

of the efficiency of any machine as a part of a plant is as largely dependent on the management as upon the design of the machine.

31 But confining our attention particularly to questions of design, we note another field of usefulness for this method of analysis. It affords a means for determining the beneficial effects of motor equipment

on the efficiency of the machine. Given any particular machine with certain possible speed-torque combinations, what changes can be wrought by the use of direct motor drive when the motor has certain characteristics of speeds and torques? The limits of this paper will not permit a full discussion of this question, but it is pointed out as one way of determining the effect of motor drive on efficiency which will lead to more definite conclusions than any number of photographs illustrating the neater appearance of a motor-equipped machine over a belt-driven machine.

32 In conclusion it may be remarked that there seems to be need for a more rational method of procedure in determining the speed-torque characteristics of a lathe. While it is impossible to formulate all the conditions which a lathe may encounter in its operation, at the same time it is believed that a method of analysis such as that described in this paper will materially assist the designer in determining the speed-torque relations which are justifiable, and will enable the purchaser to determine whether or not the speed-torque characteristics of any given lathe are adaptable to his conditions.

FINISHING STAY-BOLTS AND STRAIGHT AND TAPER BOLTS FOR LOCOMOTIVES

By C. K. Lassiter,[1] Richmond, Va.

Non-Member

The locomotive boiler of average size contains about 1500 staybolts, the number varying from 1200 in the smaller sizes to 2000 or more in the heavier types. They vary in length from $4\frac{1}{2}$ in. to $10\frac{1}{2}$ in. for the water-space bolts, which constitute about 75 per cent of the total number, to about 28 in. for the radial and crown bolts.

2 Probably no part of the boiler is subject to more destructive conditions than these little staybolts. The most serious strains are those due to expansion and contraction of the inner sheet, which bend the bolts and cause them to break close to the outer sheet. This is especially true of the side or water-space stays, which are comparatively short and have very little flexibility.

3 The material used is a high grade of refined iron, close-grained and tough. The pitch being very important on account of entering the second sheet, these stays were formerly cut to length from the bar, drilled for centers, and threaded on engine lathes. The center-drilling was not always concentric and considerable time was required to center the rough bolt so that a good thread could be obtained. This method proving too expensive, bolt cutters were used for the work, but the results were not entirely satisfactory. It was difficult to cut the threads full and smooth with one passage of the chasers and the second passage was taken at the sacrifice of pitch, as well as of time, because there was not enough material to remove to carry the chasers along properly. The introduction of the lead screw in bolt cutters brought about a very considerable improvement in pitch, but still there was trouble in getting the thread smooth for the reason

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June, 1910. All papers are subject to revision.

[1] Mechanical Superintendent, American Locomotive Company.

that the chasers were not always as accurate as the lead screw, under which conditions the threads would be rough or torn.

4 About thirty years ago the idea was conceived of concaving the bolts or reducing them in the center below the root of the thread, the object being to provide flexibility to compensate for the expansion between the inner and the outer sheets. Laboratory tests showed that a bolt reduced in the center would withstand about twice as many vibrations before breaking as one on which the threads were left straight for the full length. For many years it was the accepted practice to reduce a bolt in diameter on engine lathes after it was threaded in the bolt cutter and drilled for centers.

5 In 1900, Alonzo Epright, an engineer in the employ of the Pennsylvania Railroad, designed machines which were fully automatic in that they made from the bar, threaded and concaved, all diameters of side stays up to ten or eleven inches in length. The author has no knowledge of the production of these machines and therefore can make no comparison of costs.

FIG. 1 SQUARE END WATER SPACE STAY (PLAIN)

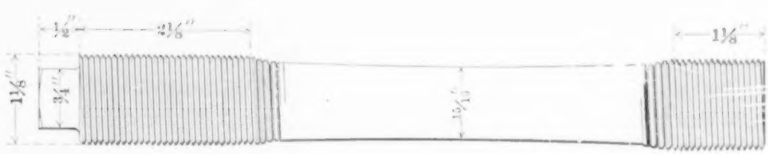

FIG. 2 SQUARE END WATER SPACE STAY (CONCAVE)

6 The vertical type of machine for threading these bolts was used to some extent and it seemed that if the proper chaser could be made the best results would be obtained from this type of machine because the weight of the head would assist the chaser to give an accurate pitch. In the horizontal or bolt cutter type the chaser must carry along the vise and carriage to the detriment of accuracy in the lead. Also, the flow of oil would assist in washing away the chips, which were troublesome in the horizontal machine. Furthermore,

the vertical type of machine is more convenient to operate, one man attending six or eight spindles with ease.

7 After a great deal of experimenting a die head was developed in which, with chasers properly ground, the limit of accuracy of 0.01 in. in 8 in. can be maintained without the use of the lead screw, which is more nearly a perfect pitch than many staybolt taps in daily use. Where a proper lubricant is used a very fine, smooth thread can be obtained at a uniform cutting speed of 20 ft. per min.

8 The turning or reducing tools are shown in Fig. 3, the cutting points being visible at the center, back of the chasers. To these tools

Fig. 3 Die Head for Threading Staybolts

are attached the crossheads KK, which are actuated by profilers or formers passing through the spaces LL, over which the head is drawn by the chaser, the staybolt acting as a lead screw.

9 The staybolt-threading machine is shown in Fig. 4. The several die heads are attached by small rods to straps passing over the pulleys on a shaft at the top of the machine. The operator grasps one of the strap handles with his right hand and, by the aid of the rotating

pulley over which the strap passes, raises the die head until it comes in contact with the bracket which closes the die. With his left hand he places the squared end of a staybolt in a holder underneath the die and allows the head to drop until the chasers begin to cut, when he moves to the next die head and repeats the operation. By the time he has placed all the heads in operation, the first bolt is finished, the die having dropped automatically when the threading was completed.

FIG. 4 STAYBOLT-THREADING AND REDUCING MACHINE, WITH SPECIAL GRINDER FOR CUTTING TOOLS

10 In Fig. 4, the die head at the right is shown raised sufficiently to insert the staybolt in place; the next at the left is just beginning to thread the bolt and the two other die heads are in still lower positions.

11 A comparison of costs by the two methods, taking a $7\frac{1}{2}$-in. side stay as an average length, would be about as follows:

FORMER PRACTICE

Threading-in bolt cutter, usually taking two cuts at 20 cents	$0.40
Drilling for centers	0.22
Concaving or reducing on engine lathe	0.75
Cost per hundred	$1.37

PRESENT PRACTICE

Present cost, threaded the entire length or threaded and concaved for all sizes and lengths, per hundred	$0.13

Using the average number of stays, a saving of labor cost of $18.60 per boiler is obtained with a minimum of rejected stays.

METHODS OF DRILLING STAYBOLTS

12 The telltale holes which are drilled in the staybolts have been the cause of considerable expense and annoyance. Some railroads drill them after the stays are placed in the boiler, with pneumatic hand drills. Under these conditions there is danger that the hole may not be central. It often happens that the drill runs through into the water space or is broken off in the hole. In either case it is necessary to remove the bolts and put in others. Sometimes the holes are drilled on a vertical drilling machine before being placed in the boiler. Even then the breakage of drills is very large, averaging about sixteen to the boiler, and each broken drill means a staybolt thrown away.

13 An automatic machine has been devised for drilling these holes before the stay is placed in the boiler. They are fed from a hopper and automatically centered in position for the drill. When the hole is bored about one-third of the depth, the drill is withdrawn and the bolt is carried forward in the turret mechanism which holds it to a second and a third drill, completing the hole. Each drill is 0.01 in. smaller than the preceding one, providing for a minimum of friction and a maximum of clearance for chips. The holes are of uniform depth and in the center of the bolt. The average breakage is about three or four drills to the boiler.

COMPARISON OF COSTS

Drilling in the boiler, per hundred (to which should be added the cost of replacements	$0.90
Drilling under drill press, per hundred (to which should be added cost of drills and waste of material and labor)	0.45
Drilling in the automatic machine, per hundred (with the minimum number of broken drills and bolts destroyed)	0.12

METHODS OF FINISHING STRAIGHT AND TAPERED BOLTS

14 The usual method of finishing straight and tapered bolts for locomotives was to drill for centers, place in engine lathes, face under the head, turn the body taper, turn the part to be threaded straight and to proper size, face down the thread end to length and shape, leaving the center intact, test and file to accuracy, and cut off center point, after which the bolt is ready to be threaded in the bolt cutter and to have the hexagon head changed to any special shape desired.

15 About 1889, S. M. Vauclain, Mem.Am.Soc.M.E., designed and used a turning head in connection with a vertical machine for bolts up to 12 in. long. Under rights obtained from him the Pennsylvania Railroad placed an equipment of this kind in its Altoona shops and that is the only railroad known to the author using other than engine lathe methods in finishing bolts.

16 As a great many straight and tapered bolts used in locomotives are 12 in. to 20 in. in length and even longer, it became necessary to design for this work a turning head which would handle taper bolts up to 18 or 20 in. in length and up to $1\frac{3}{4}$ in. diameter of thread, and straight bolts in any length up to 27 in. and up to $2\frac{1}{2}$ in. diameter. It may be quite possible to go beyond these dimensions should the specifications require. These requirements have been met by a special machine of the vertical, multiple-spindle drill type, with which is used a special cutter head shown in Fig. 5. This head is the real or essential means of producing these bolts, either straight or taper and cylindrically true to the axis, the machine being simply a proper means of driving and feeding the bolt during the turning operation.

17 The cutter head consists of a retaining shell of cast iron, the bore of which must be round and straight; six segments, three of which are rigidly fastened to the shell, the other three having a limited amount of freedom and being fastened in place by a taper key with an adjusting screw located in the center of the radius with a bearing on the shell; and three blades, alternating with three guides, placed between the segments and backed up with taper keys and adjusting screws. The taper keys, in connection with a certain amount of taper on the blades and guides, have sufficient movement to provide for about one-eighth inch adjustment for re-grinding of the blades, or with the same amount on the guides, one-quarter inch in diameter of bolts. It will readily be seen that when an accurately ground plug gage of the size that it is desired to turn the bolt is placed centrally in the head, the blades and guides can be adjusted to their proper position. The three

loose segments are then forced forward by the taper key, clamping the blades and guides rigidly in their proper working position.

18 The economical use of this method of turning bolts, particularly in the railroad shops and locomotive works where taper bolts are largely used, necessitates a change of system. The usual practice, especially on repair work, has been to carry in stock only standard sizes of forgings, though in some cases the more common sizes were placed in stock finished. With the engine lathe located near the loco-

FIG. 5 CUTTER HEAD AND ATTACHMENTS

motive being repaired, the bolts were fitted to the hole after the least possible amount of reaming had been done that would clean up the hole.

19 The improved system contemplates the turning, facing under the head, and placing in stock of standard sizes in lengths of 6, 9, 12, 15, and 18 in. and varying in diameter under the head by thirty-seconds of an inch. Stock may be kept in sixty-fourths of an inch if desired, but very few holes will be found which require less than thirty-

seconds of an inch to clean up. In fact, the chief reason for carrying the intermediate sizes would be to save the hole when it cannot be cleaned up within the next thirty-second. Standard reamers are used, with collars or marks to indicate when they have been driven to the required depth. All bolts have standard hexagon heads conforming to the thread diameter.

20 Bolts are specified with relation to the length and the diameter under the head, and the stock size next longest is used. Under these conditions not more than 3 in. must be cut off to bring the bolt to the proper length. The stock bolts are then taken to the bolt-altering machine, which is a quick-acting hand machine equipped with collet chucks and split bushings for the various diameters of the bolts. The end may be cut off to the proper length and turned for cotter pins, and the head changed to counter sink, box head, button head, or whatever may be required. After threading on the bolt cutter, the bolt is ready to drive in place without further fitting.

21 A comparison of costs by the two methods, taking a $1\frac{1}{8}$ in. × 9 in. bolt as an average would be about as follows:

ENGINE LATHE PRACTICE

	Cost per hundred
Drilling for centers	$0.22
Turning in lathe	2.50
Altering in lathe	$2.50 to 3.50
Threading in bolt cutter	0.22
Cutting off center points	0.10

PRESENT PRACTICE

Pointing the blank	$0.12
Turning by the method described	0.45
Cutting off and changing points and heads where necessary on the bolt-altering machine	$0.40 to 0.60
Threading in the bolt cutter	0.22

22 A device is now being perfected by which the threading can be done automatically at the same time the turning is done. This not only eliminates the bolt cutter charge of $0.22 per hundred, but assured a full, uniform thread absolutely in line with the body of the bolt and square with the facing under the head. When used in connection with a nut faced square with its thread the most satisfactory bolt is obtained.

23 A combined turning and threading device implies a modified form of the cutter head previously described, underneath which is

TABLE OF STOCK SIZES

SHOWING EIGHT THREADED DIAMETERS OF BOLTS AND THIRTY-TWO DIAMETERS UNDER THE HEAD

Thread Diameter	$\frac{3}{4}$				$\frac{7}{8}$				1				$1\frac{1}{8}$			
Diameters under head	$\frac{25}{32}$	$\frac{13}{16}$	$\frac{27}{32}$	$\frac{7}{8}$	$\frac{29}{32}$	$\frac{15}{16}$	$\frac{31}{32}$	1	$1\frac{1}{32}$	$1\frac{1}{16}$	$1\frac{3}{32}$	$1\frac{1}{8}$	$1\frac{5}{32}$	$1\frac{3}{16}$	$1\frac{7}{32}$	$1\frac{1}{4}$
Length under head	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
					9	9	9	9	9	9	9	9	9	9	9	9
										12	12	12	12	12	12	12
													15	15	15	15
															18	18

Thread diameter	$1\frac{1}{4}$				$1\frac{3}{8}$				$1\frac{1}{2}$				$1\frac{5}{8}$			
Diameter under head	$1\frac{9}{32}$	$1\frac{5}{16}$	$1\frac{11}{32}$	$1\frac{3}{8}$	$1\frac{13}{32}$	$1\frac{7}{16}$	$1\frac{15}{32}$	$1\frac{1}{2}$	$1\frac{17}{32}$	$1\frac{9}{16}$	$1\frac{19}{32}$	$1\frac{5}{8}$	$1\frac{21}{32}$	$1\frac{11}{16}$	$1\frac{23}{32}$	$1\frac{3}{4}$
Length under head	6	6	6	6	6	6	6	6								
	9	9	9	9	9	9	9	9								
	12	12	12	12	12	12	12	12	12	12	12	12				
	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15
	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18

attached a die head of special construction. This die head is carried on four or more vertical rods or guides which are attached to a ring to which the cutter head is fastened. Provision is made for squaring the die head with the cutter head at the time it begins cutting the thread, and at the same time automatically placing the die head in a position where it is free to move in a vertical plane up or down in exact proportion to the difference between the feed and the pitch of the thread to be cut. An automatic knock-out is provided which opens the die head and passes to one side, allowing the threaded bolt to go through to any length within the feed of the machine. Under these conditions it will be seen that so long as the length of the thread to be cut is the same, the length of bolt to be turned is immaterial. The device is very simple in its construction and does not call for a skilled mechanic to adjust or operate it.

TWO PROPOSED UNITS OF POWER

By Prof. Wm. T. Magruder, Columbus, O.

Member of the Society

James Watt is said to have defined a "horsepower" as 33,000 foot-pounds of work per minute, and a "boiler horsepower" as the evaporation of a cubic foot (62 lb.) of water per hour. His rule is sometimes put into the form that "one square foot of grate surface, one square yard of heating surface, a half of a square yard of water surface, and one cubic yard of contents, equals one horsepower, and will evaporate one cubic foot of water per hour in a waggon boiler."

2 Charles E. Emery, Charles T. Porter and Joseph Belknap, "Committee on Boiler Trials of the Judges of Group XX," reported through Horatio Allen, Chairman of Group XX, to Prof. Francis A. Walker, Chief of Bureau of Awards of the United States Centennial Commission of the International Exhibition of 1876, that "the estimated Horse-Power of the several boilers" was given "on the basis that the evaporation of thirty pounds of water is required per horse-power per hour, the results being derived from evaporation at steam pressure of 70 pounds from temperature of 100°."[1] In the Report of the Committee of Judges of Group 20, p. 131, as published by J. B. Lippincott & Co., Philadelphia, "the commercial horse-power of a boiler is fixed at 30 pounds of water evaporated at 70 pounds gage pressure from a temperature of 100 deg."[2] It is to be noted that the time element is omitted. This is not an unusual mistake in speaking of rates, the time element being understood, or taken for granted. This definition is commonly modified so that the Centennial standard of horsepower or the "Centennial horsepower" is defined as the

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

[1] United States International Exhibition, 1876. Reports and Awards, vol. 6, p. 426, sect. 35.

[2] Trans. Am. Soc. M. E., vol. 21, p. 84. Report of Committee on Revision of Standard Code.

"evaporation of thirty pounds per hour of water at 100 deg. fahrenheit into dry steam at 70 pounds gage pressure, or the equivalent." This last phrase is very generally omitted. It is interesting to note that the committee called it the "commercial horsepower," and not a "boiler horsepower." They also defined the "unit of evaporation" as one pound of water at 212 deg. fahr. evaporated into steam of the same temperature, and as being equivalent to 965.7 heat units.

3 In the Appendix is given a summary of the reports of committees of the Society relating to horsepower units and of various discussions on the subject. From these it will be seen that while trying to keep to the Centennial standard of commercial (boiler) horsepower, the committees have gradually veered to the thermal-unit standard, and from the standard of 30-lb. from 100 deg. fahr., into steam at 70-lb. gage pressure.

4 Since 1876, the accuracy of our knowledge of the heat of steam has increased. This is especially true since 1899, the time of the last report to the Society. The "confusion to practical boiler owners," which Dr. Chas. E. Emery seemed[3] to fear might result from the practice of measuring the power of a steam boiler in heat units, does not seem to have materialized.

5 The publication of the new eighth edition of Professor Peabody's Tables of the Properties of Steam, and the publication of the Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam by Professors Marks and Davis, have complicated this matter still more, and especially with engineering students.

6 According to the steam tables of Charles T. Porter, and the various reports that have been referred to on this subject, the value of the "unit of evaporation" is 965.7 B.t.u. According to Peabody, its value has been gradually changing to 965.8, 966.3, and now to 969.7. According to Marks and Davis, its value should be 970.4. These differences amount to only 4.7 B.t.u. in 970.4, or to one in 205, which is one-half of one per cent. It would seem desirable to use 970 hereafter, instead of 966, as the unit of evaporation, this being the average of the most accurate determinations of the latent heat of evaporation of water at 212 deg. fahr.

7 Similarly, the "unit of commercial evaporation" has been changing from 1110.2 B.t.u. in 1876 and 1884, to 1115.0 according to Peabody, and to 1115.6 B.t.u. according to Marks and Davis today.

[3] Trans. Am. Soc. M. E., vol. 6, p. 334. Report of Committee on Revision of Standard Code.

8 When measured in thermal units, the value of the boiler horsepower, 34½ units of evaporation, is given as 33,305 B.t.u. by the Centennial judges and by the committee reporting to the Society in 1884; as 33,317 B.t.u. in the report of the committee as made in 1899; as 33,320 B.t.u. in one text book on steam-boilers; as 33,454.7 B.t.u. (34½ x 969.7) by Peabody; and as 33,478.8 B.t.u. (34½ x 970.4) by Marks and Davis.

TABLE 1 DIFFERENT VALUES OF A BOILER HORSEPOWER IN B. T. U.

	Units of Evaporation		Units of Commercial Evaporation		B. T. U.
	One	34½	One	30	
Centennial	965.7	33,317	1110.2	33,306	33,305
Peabody	969.7	33,455	1115.0	33,450	....
Marks and Davis	970.4	33,479	1115.6	33,468	....

9 It must be evident to everyone that a would-be standard which has so many different thermal values and is capable of acquiring others with each change in the steam tables is not only indefinite but confusing. It is not a definite unit of measurement, which all standards should be. It seems a pity that in the definition of such a commonly used engineering term there should be any possible chance for confusion and misunderstanding on the part of the student, or for litigation between contractors over the accuracy of the fulfillment of the terms of the contract.

10 Again, for over thirty years, engineers and engineering teachers have been apologizing for the use of the term "boiler horsepower." Even the committee of the Society which reported in 1884, says,[4] "It cannot properly be said that we have any natural unit of power for rating steam boilers." If a horsepower is the rate of doing work, and a boiler is considered as a machine, and the water as the moving parts, the only mechanical power that a boiler produces is that due to the external latent heat of evaporation, except when it explodes. Hence the term "boiler horsepower" is a misnomer. The object of the use of a boiler is the absorption of the heat energy obtained from the potential energy of the fuel by combustion, and the transfer to and

[4] Trans. Am. Soc. M. E., vol. 6, p. 263. Report of Committee on Revision of Standard Code.

storage of the same by a volatile liquid for convenient use in a heat engine, or for other thermal purposes. Hence as a boiler uses the latent heat energy of the fuel as its source of supply, and develops and delivers available heat energy, there would seem to be every reason why the power or ability of a boiler to deliver energy should be measured in thermal units, as being the only unit of energy that the boiler ever normally receives or delivers. Furthermore, the energy from every boiler is always measured in heat units before being reduced to boiler horsepower.

11 To measure the capacity or power of a boiler plant, or its output of energy, in millions of thermal units would not be practical; a smaller unit is desirable. It is therefore proposed to measure the power or capacity of a boiler in "boiler-powers," and to define a boiler-power as 33,000 B.t.u. of heat energy delivered per hour by a steam-boiler, steam main, or by a hot-water heating main, or the like, or added per hour to the feed-water of a boiler, or to the water of a hot water heating system. The acceptance of this term will, it is thought, simplify the whole subject; the unit will remain constant, will be easily remembered and easily used, and will not be one of three standards, differing slightly among themselves, as is at present the case with the term boiler horsepower. Its analogy to mechanical horsepower will be helpful rather than the opposite, especially to the beginner in engineering knowledge. The unit boiler horsepower may still be retained by those who may prefer to use it in some one of its many thermal values.

12 The rapid introduction of gas engines using blast furnace, coke oven, or producer gas, leads to the suggestion of a new unit for the capacity or power of a gas producer, coke oven, or blast furnace, to deliver available heat energy for use in gas engines, under stoves and boilers, or for other thermal uses.

13 At the St. Louis meeting of the American Association for the Advancement of Science in December 1903, the writer read a paper suggesting the term "producer horsepower" as a unit. Since then the question has arisen as to why the old misnomer of "horsepower" should be perpetuated as a unit of measurement of heat energy. Why not simplify and shorten the term "producer horsepower" to "producer power?" If such a unit is desirable for the measurement of the capacity or power of a gas producer, why not suggest similar ones for other generators of heat energy available for use in gas engines and for other thermal uses? Instead of measuring the power of a gas producer in producer powers, and the powers of a blast furnace and

of a coke oven to generate heat energy in blast-furnace powers and coke-oven powers, it is proposed to include all such sources of power, and to measure the heat energies of gaseous and liquid fuels, in "gas powers," and to define a gas-power to be 10,000 B.t.u. of heat energy delivered per hour by a gaseous or liquid fuel. The calorific value should be measured from and to 62 deg. fahr., and at 30 in. of mercury. This unit can be applied and used in the measurement of the energy delivered by a gas well, a gas main, a gas producer, a blast furnace, a coke oven, an oil well, or a pipe line.

14 The number, 10,000 B.t.u., has been chosen as the average in the best gas-engine practice today of heat energy required to develop a horsepower of mechanical energy. The figure bears to current gas-engine practice about the same relation that 30 lb. per hr. of steam at 70 lb. gage pressure from water at 100 deg. fahr. did to current steam-engine practice in 1876. The definition as given contemplates using only the higher calorific value of the fuel, rather than the lower or, so-called, effective value.

15 It is to be hoped that some such unit for the measurement of the output of a generator of heat energy in gaseous or liquid form can be found, and adopted by common consent, before practice and commercial custom in different portions of the country shall have learned to use units which have been less carefully selected and less accurately defined.

APPENDIX

In a paper presented before the Society by Wm. Kent,[1] at the Pittsburg Meeting in May 1884, he tabulates the "horsepower developed at 30 lb. of water evaporated per hour from and at 212°. (Page 268)." In a footnote we read, "The customary method of rating horsepower is 30 lb. of water per horsepower per hour from a feedwater temperature of 212° into steam at 70 lbs. pressure above the atmosphere, which is equal to 30.985 lbs. from feed at 212° into steam of the same temperature. The writer prefers the calculations both of economy and horsepower to be made on the basis of evaporation from and at 212°, for the sake both of uniformity and of convenience in calculation."

2 In a paper presented before the Society by Dr. Chas. E. Emery at the same meeting,[2] he defines the Centennial horsepower (C.H.P.), as being "thirty Kals per hour;" and a "Kal" as being "one pound of water evaporated into saturated steam at seventy pounds pressure from temp. of 100°, with a thermal value of 1110.2 thermal units (Page 282)." This would make a Centennial horsepower equivalent to 33,306 B.t.u. In discussing this paper, Mr. Kent argued that "in determining the horsepower in a steam boiler . . . we should start with the British thermal unit as a basis . . . The unit of evaporation should be 965.7 thermal units. It is the evaporation of one pound of water from and at 212 degrees. A horsepower should be a definite number of units of evaporation—say 30 (Page 297)." He was followed by Dr. E. D. Leavitt, Jr., who thought that "the simplest proposition was to come down to thermal units." Dr. Emery assented that any unit proposed must be based on the "heat unit." In the Centennial Report, this amount of heat energy (1110.2 B.t.u.) was termed the "unit of commercial evaporation (Page 300)."

3 In the Report of the Committee on a Standard Method of Steam-Boiler Trials made at the New York Meeting of the Society in November 1884,[3] the Centennial unit of boiler-power is stated as "30 pounds of water evaporated into dry steam per hour from feed-water at 100° Fahrenheit, and under a pressure of seventy pounds per square inch above the atmosphere." "The quantity of heat demanded to evaporate a pound of water under these conditions is 1110.2 British thermal units, or 1.1496 units of evaporation. The unit of power proposed is thus equivalent to 33,305 heat-units per hour, or 34.488 units of evaporation." Another standard unit for the power of a boiler which was suggested to the Committee in 1884 was "the evaporation of thirty pounds of feed-water into dry steam from and at the boiling point at mean atmospheric pressure (212°F.) (Page 265)." This would have then been taken as the equivalent of 30 x 965.7 = 28,971 B.t.u. per hr. It was not accepted.

[1] Trans. Am. Soc. M. E., vol. 5, p. 260. Rules for Conducting Boiler Tests.

[2] Trans. Am. Soc. M. E., vol. 5, p. 282. Estimates for Steam Users.

4 This Committee recommended in 1884 the adoption of the Centennial Standard and that, for standard trials of steam boilers, the "commercial horsepower be taken as an evaporation of 30 pounds of water per hour from a feed-water temperature of 100° Fahr. into steam at 70 pounds gauge pressure, which shall be considered to be equal to 34½ units of evaporation; that is, to 34½ pounds of water evaporated from a feed-water temperature of 212° Fahr. into steam at the same temperature. This standard is equal to 33,305 thermal units per hour (Page 266)." A footnote gives the "evaporation of 34½ pounds from and at 212°F., as being equal to 30.010 pounds from 100°F., into steam at 70 pounds pressure," and "the 'unit of evaporation' as being 965.7 thermal units," according to the tables in Porter's Treatise on the Richards Steam Engine Indicator, which was the standard of that day.

5 Dr. Chas. E. Emery stated[3] that the "commercial horsepower of 34½ units of evaporation per hour is, for all practical purposes, equal to 33,333 thermal units per hour making it convenient to obtain the horsepower by multiplying the total number of thermal units derived from the fuel per hour by 0.00003 (Page 304)."

6 Prof. J. B. Webb in speaking about the definition of a "commercial horsepower" said "To my mind it would be simpler and better to express results in *thermal units per hour*, and at all events not to express them in horsepowers which are very far from being horsepowers (Page 322)." Nothing came of his suggestion.

7 During this discussion, Wm. Kent introduced and used the term "boiler horsepower" rather than "commercial horsepower," and quoting the definitions of commercial horsepower as recommended by the committee, added, "This standard is certainly not open to the charge of want of exactness and precision (Page 324)."

8 The Committee in its report said that it had "concluded to recommend thirty pounds as the unit of boiler-power (Page 264)".

9 Dr. Charles E. Emery stated that "it was informally suggested to make the standard exactly 33,000 British Thermal Units per hour, so that it would be numerically the same as the number of foot-pounds per minute constituting an actual horsepower, and again 33,333 B.t.u. were suggested to facilitate the calculations, but the general feeling of the committee was against any change whatever (Page 333)." He adds, and seems to prefer the statement that "The value of the unit of horsepower announced is 33,305 British Thermal Units per hour, which being stated in the Report definitely fixes the standard. It also equals 34½ units of evaporation, within one-thirtieth of one per cent."

10 Prof. W. P. Trowbridge, in discussing the report, called attention to the diversity of opinion in the committee as to whether the "unit of boiler-power" should be expressed in terms of the "unit of evaporation," or in some other terms.

11 Prof. W. P. Trowbridge and Prof. C. B. Richards presented a paper at the Boston meeting in 1885 on The Rating of Steam Boilers by Horse-Powers for Commercial Purposes,[4] in which they differ from the committee which had reported the preceding year, but quote its report with the statement "What is

[3] Trans. Am. Soc. M. E., vol. 6, p. 256.
[4] Trans. Am. Soc. M. E., vol. 7, p. 214.

needed is a standard unit of boiler power which may be used commercially in rating boilers, and in specifications presenting the power to be demanded by the purchaser and guaranteed by the vender (Page 216)."

12 George H. Babcock in discussing this subject said, "A dynamic horsepower in its simplest form is 33,000 foot-pounds per minute. A boiler horsepower should be defined as 33,000 heat units per hour imparted to the water (Page 225)." Mr. Kent stated, "The term horsepower has two meanings in engineering literature: First, an absolute unit or measure of the rate of work, ... and an approximate measure of the size, capacity, value, or "rating," of a boiler engine, water-wheel or other source or conveyor of energy, by which measure it may be described, bought and sold, etc. (Page 226)."

13 In the Report of the Committee on the Revision of the Standard Code for Conducting Steam-Boiler Trials, made in 1899,[5] it is stated (Page 36) that "The Committee approves the conclusions of the 1885 Code to the effect that the standard 'unit of evaporation' should be one pound of water at 212 degrees Fahr. evaporated into dry steam of the same temperature. This unit is equivalent to 965.7 British thermal units. The Committee recommends that, as far as possible, the capacity of a boiler be expressed in terms of the 'number of pounds of water evaporated per hour from and at 212 degrees.' It does not seem expedient, however, to abandon the widely recognized measure of capacity of stationary or land boilers expressed in terms of 'boiler horsepower.' The present committee accepts the same standard, but reverses the order of the two clauses in the statement, and slightly modifies them to read as follows: 'The unit of commercial horsepower developed by a boiler shall be taken as 34½ units of evaporation per hour; that is, 34½ pounds of water evaporated per hour from a feed-water temperature of 212 degrees Fahr. into dry steam of the same temperature. This standard is equivalent to 33,317 British thermal units per hour. It is also practically equivalent to an evaporation of 30 pounds of water from a feed-water temperature of 100 degrees Fahr. into steam at 70 pounds gauge pressure.'" In a footnote is added the statement that "The unit of evaporation being equivalent to 965.7 thermal units, the commercial horsepower = 34.5 x 965.7 = 33,317 thermal units (Page 37)."

[5] Trans. Am. Soc. M. E., vol. 21, p. 34.

GAS ENGINES FOR DRIVING ALTERNATING-CURRENT GENERATORS

H. G. Reist, Schenectady, N. Y.

Member of the Society

The problem of driving an alternating-current generator by means of a gas engine is not inherently different from that of driving it from a steam engine. If the shaft of the engine turned with a uniform motion, no difficulty would be experienced and no special design would be necessary. It is the variations in angular velocity and speed that affect the driving of alternators.

2 If the current of a single generator is used for lights or for heating, as in electric furnaces or in electrolytic work, variations in velocity either during a single turn or due to the hunting of the governor will simply increase and decrease the load as the speed varies. If induction motors are driven from a single generator, it is only under peculiar circumstances that any trouble is experienced due to speed variations in the engine, because this type of motor is asynchronous and does not have to follow exactly; it is as if it were belted to the engine, the connection being slightly flexible.

3 A synchronous motor or rotary converter, on the other hand, must keep in phase and behave as though geared to the engine, and must respond to all its speed variations. If it does not keep absolutely even, that is, if its phase relations change, cross currents will flow. When two or more generators are operated in parallel, their behavior is similar, any angular departure of one from the other causing a cross-current. The volume of the cross-current depends, with any given design of generator, on the angular departure of the generators from each other. This departure may be twice the angular variation of the engine rotating parts from a mean position, because one may be a maximum distance ahead while the other is in the most backward position.

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

4 If the generators were mounted on the shaft so that the relations of the poles to the cranks were identical, and if it were possible so to synchronize them that the corresponding cranks of engines to be run together were exactly together, no cross-currents would flow, because the engines would slow down and speed up together. This is not feasible, however, and it becomes necessary to design the engine to run with a fairly uniform rotation. It has been found good practice to limit the variation from a mean position of the revolving parts of the electric generator to $1\frac{1}{4}$ electrical degrees.

5 An electrical degree is 1-360 part of the space occupied by two poles on a generator; that is, a two-pole generator is the unit and an electrical degree is one mechanical degree of such a machine; if the generator has four-poles, an electrical degree will be one-half a mechanical degree of the circle on which the poles are mounted; if there are 6 poles, it will be one-third of a mechanical degree. In general, to reduce electrical degrees to mechanical degrees, we must divide the allowable variation by one-half the total number of poles on the generator; so that the $1\frac{1}{4}$ electrical degrees mentioned above for a twenty-pole machine would be 0.125 actual degrees on the circumference of the flywheel. From this it is evident that with a generator of many poles, a more even speed is needed than for one with few poles. For a 60-cycle generator, which at a given engine speed has $\frac{60}{25}$ as many poles as a 25-cycle generator, the evenness of running must be much greater than for a 25-cycle generator.

6 The cross-currents between two electrical generators tend to speed up the lagging machine, bringing them more closely into synchronism. If there were no inertia the rotating parts of generator and flywheel would quickly get into synchronism, reducing and almost eliminating the cross-currents. This is, however, an ideal condition. The cross-currents are a factor of the amount of inertia with a given natural angular variation, and it will readily be seen that from this standpoint the larger the flywheel the less effect a given value of currents or torque will have on the mass. If the flywheel is very large, the currents which it may be practical to allow to flow between the machines may not be able to draw them together at all. Hence a large flywheel, while useful in obtaining uniform rotation, so far as the engine is concerned, prevents the current flowing between the machines from being very effective in drawing them into synchronism. This shows that it is desirable to obtain uniform rotation in other ways than by the use of an excessively heavy flywheel. Currents flowing between machines occasion losses in the copper and this

adds to the heating of the machine. They thus reduce the output with a given rise of temperature and reduce the efficiency.

7 Certain elements of design may be introduced into an electrical generator to make it less sensitive to slight variations in turning moment supplied by the engine, such as building a generator of poor regulation. The regulation must not be too poor, however; otherwise the operation of the system will be unsatisfactory. A "squirrel-cage" winding in the poles of the generator allows secondary currents to flow in this part of the structure and increase the torque, tending to draw the generators together with a given interchange of current between the two machines. This is of great assistance in parallel operation of generators and should generally be applied on generators to be driven by gas engines. If a flexible connection could be provided between engine and generator it would greatly assist in satisfactory parallel operation, but this connection is generally applicable only on small machines. The ultimate solution lies in the direction of greater uniformity of motion in the engine itself.

8 Uniformity of rotation of gas engines is dependent on a number of elements of design, such as (*a*) the number of impulses per revolution, which in turn is dependent on the number of cylinders and arrangement of cranks and on whether a two or a four-cycle system is used; (*b*) the compression and weight of the reciprocating parts; (*c*) the time of ignition; (*d*) the weight of the flywheel. The use of a heavy flywheel, however, while one of the simplest, is the least desirable method of obtaining even rotation of the engine shaft, and other means should be used to obtain as uniform rotation as possible.

9 The following seem to be the desirable characteristics of gas engines for driving alternators:

- *a* High speed. This will require fewer poles with a given frequency and a greater angular variation will be allowable.
- *b* A light flywheel. This will allow the current to keep the generators together with a minimum disturbance.
- *c* Large engines should be built with many cylinders and cranks so placed as to contribute to an even turning moment.

CRITICAL SPEED CALCULATION

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Non-Member

Critical speed is the term applied to the speed of a rotating body at which occur the maximum vibrations of the revolving mass or supporting structure. The vibrations are smaller for speeds both above and below the critical value. Hence the importance, to the designer of high rotative speed apparatus, of predetermining these maximum vibrating points. The high speeds and large capacities now being used in electrical machinery, such as turbo-generators, frequency changers, etc., bring this apparatus within the critical-speed range; and the electrical designer must study the vibrating properties of his high speed machines, or leave this operating trouble to chance.

2 The phenomenon of critical speed was known to De Laval, who designed his turbines with a small or "flexible" shaft, so that the running speed was seven to ten times the critical value. So far as is known he did not understand the mathematical theory.

3 The first scientific explanation of critical speed is due to Rankine who in Machinery and Millwork gave the mathematical explanation for a shaft with its own weight only, when supported at each end, and also for fixed direction at one end as a cantilever. This was followed by Professor Greenhill[2] with an explanation for an unloaded shaft with fixed direction at each end. Professor Reynolds[3] then extended the mathematical treatment to shafts loaded with pulleys, and Professor Dunkerley[3] proved the formulæ by laboratory experiments and developed an approximate formula for shafts with more than one load. Reynolds and Dunkerley do not satisfactorily treat

[1] General Electric Company, Schenectady, N. Y.

[2] Proceedings Institution of Mechanical Engineers, April 1883.

[3] Philosophical Transactions, Royal Society, London, vol. 185a, 1895; Proceedings, Liverpool Engineering Society, 1895.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

the case of two loads on the shaft and their method was criticised by Dr. Chree.

4 Föppl[1] in Germany gave the case for a single concentrated load of the shaft. Stodola[2] in 1903 first gave the formula for any two concentrated loads on a shaft. Professor Morley[3] has lately given approximate formulæ for combined distributed and concentrated loads.

5 These constitute practically all of the literature on the subject. They are mainly mathematical demonstrations and do not leave the subject in convenient form for the use of the designing engineer. This paper will give a mathematical treatment for both the distributed and the concentrated loads, by considering the motion of the shaft as vibratory along two axes, study the vibrations for all speeds, reduce the formulæ to practical form, and present them in tables for convenient use.

NATURE OF CRITICAL SPEED

6 To explain more easily the nature of critical speed we will first give the simple solution of Föppl for a single load. All critical-speed calculations assume an unbalanced load. It is practically impossible to balance a rotating mass so that its center of gravity exactly coincides with the mechanical axis of rotation. As the mass starts to rotate, the center of gravity will rotate in a very small radius around the shaft center. The rotation of the center of gravity at this small radius produces a centrifugal force which acts radially outward from the shaft center through the center of gravity, and rotates around the shaft with the center of gravity. Consider the case shown in Fig. 1, of a single concentrated load on a vertical shaft. Let a be the unknown distance from the center of gravity of the mass to the center of the shaft. The centrifugal force of this mass m, rotating at the radius a, will tend to deflect the shaft in the direction of a, so that the shaft will rotate in a bowed condition. The bowed shape will in itself increase the circle in which the center of gravity rotates; this increases the centrifugal force, and in turn the shaft deflection. This action continues until finally a state of equilibrium is reached where the force of the shaft deflection is equal and opposite to the centrifugal force of the mass. This condition of equilibrium is shown

[1] Civil-Ingenieur, 1895, p. 333.
[2] The Steam Turbine, Stodola, p. 183.
[3] Engineering (London), 1909, vol. 88, p. 135.

in Fig. 2, where the center of gravity is rotating at the radius r, and the shaft rotating in a bowed condition at the radius or deflection $(r\text{-}a)$. Let $\Delta = \frac{W\,K}{E\,I}$ — static deflection of shaft, if horizontal, and p (angular velocity) $= \frac{2\,\pi\,N}{60\,\cancel{N}}$, where N = r. p. m. The centrifugal force of the center of gravity is mrp^2. This centrifugal force would produce a deflection of $m\ r\ p^2 \frac{K}{E\,I} = r\ \ p^2 \frac{W\,K}{g\,E\,I} = r\,p^2 \frac{\Delta}{g}$, where g is gravity. But the shaft deflection opposing the centrifugal force is, for equilibrium $(r\text{-}a)$. This gives the equation

$$r - a = r\,\frac{p^2\,\Delta}{g}$$

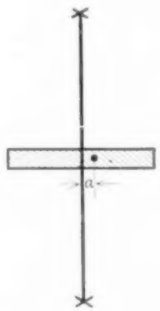


FIG. 1 CONCENTRATED LOAD ON VERTICAL SHAFT AT REST

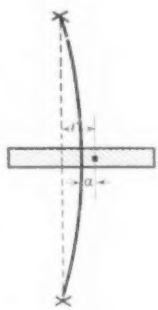


FIG. 2 CONCENTRATED LOAD ON VERTICAL SHAFT IN MOTION

which solved for r gives

$$r = \frac{\dfrac{ga}{\Delta}}{\dfrac{g}{\Delta} - p^2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

7 This equation, with values of r plotted against p, is shown in dotted lines in Fig. 3. As the angular speed p increases from zero, the radius or deflection r increases, until r becomes theoretically infinite when

$$p^2 = \frac{g}{\Delta}$$

This is the condition of maximum vibration produced by the shaft, and the critical number of revolutions is found from the equation

$$p^2 = \frac{g}{\Delta}$$

$$\left(\frac{2\pi}{60} N\right)^2 = \frac{32.2 \times 12}{\Delta}$$

$$N = \frac{187.7}{\sqrt{\Delta}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

for inch, pound, minute, units.

8 Referring to the curve beyond the critical-speed value, r becomes negative, and as the value of p is increased r approaches the

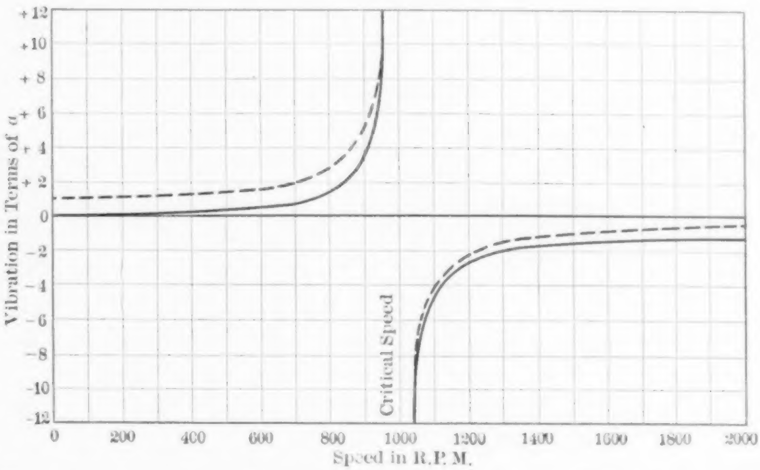


FIG. 3 AMPLITUDE OF VIBRATION WITH SINGLE CONCENTRATED LOAD (FIGS. 2 AND 4).

DOTTED LINE FOR EQUATION 1. SOLID LINE FOR EQUATION 4

limit of zero; in other words, above the critical speed the center of gravity revolves inside the bow of the shaft, or in a smaller circle than the shaft center; and the tendency of the rotating mass is to rotate about its own center of gravity, and not about the mechanical center. It approaches its center of gravity as a limit for infinite speed.

9 The natural time of vibration of a loaded shaft is

$$t = 2\pi\sqrt{\frac{\Delta}{g}}$$

and the number of natural vibrations per minute is

$$\frac{60}{t} = \frac{60}{2\pi}\sqrt{\frac{g}{\Delta}}$$

which is the same as N in Equation 2, the critical number of revolutions. Thus for a single concentrated load the critical-speed phenomena occur when the revolutions synchronize with the natural period of vibration of the shaft. No satisfactory explanation has been given of the detail action at the critical speed, or of the manner in which the center of gravity passes from the outside to the inside of the bow of the shaft. Theoretically the deflection or bow of the shaft becomes infinite at the critical speed. Practically it does not, because of the resistance of the air and probably the need of the factor of time to accumulate energy.

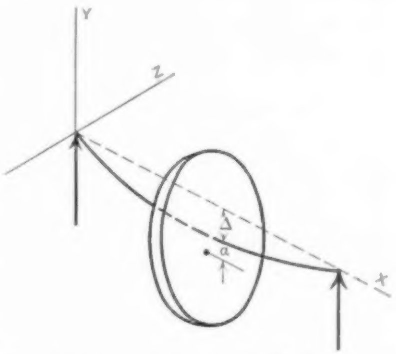


FIG. 4 CONCENTRATED LOAD ON HORIZONTAL SHAFT

10 In machines where the normal running speed is higher than the critical speed, the shaft is made just strong enough to withstand the deflection in passing through the critical speed, and as weak or flexible as possible for the smooth running above the critical speed. The weaker the shaft, the lower the critical speed, the nearer approach to rotation about the center of gravity, and the less bow or deflection in the shaft.

11 This solution is satisfactory so far as the critical value and deflection of the rotating mass is concerned; and it affords a simple explanation of the actions of a rotating body. But in the design of a machine the vibrations of the frame or supporting structure are of equal or greater importance. The shaft rotating in its bowed condition has a reaction on the bearing points, the reaction rotating with

the shaft. This force is the impressed vibration that causes the frame to vibrate. If we determine the shaft deflection during rotation, or the location of the shaft axis at any instant, we can find the amount of the force of the shaft, or the impressed vibration on the frame.

12 When coördinate axes, as shown in perspective in Fig. 4, are taken, and the location and motion of the shaft center at any instant are determined, the force impressed upon the frame is measured by the coördinates of the shaft center. If we sum the forces along each axis, the solution gives us a form of compound harmonic vibration. This same method affords a comparatively easy algebraic solution for two loads; and is applied equally well to horizontal and vertical shafts

SINGLE CONCENTRATED LOADS

13 For the condition shown in perspective in Fig. 4, Δ is the static deflection at the disc load when the shaft is horizontal, and a the distance from the shaft center to the center of gravity. To simplify the

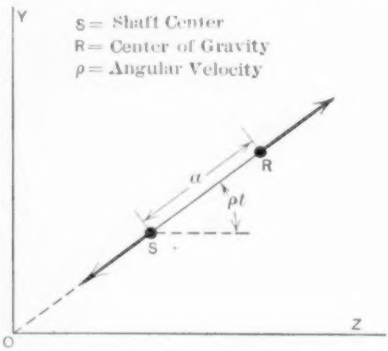


Fig. 5 Concentrated Load on Horizontal Shaft (Fig. 4)

calculations consider the shaft in a vertical position. It will be later shown that the same formulæ apply to horizontal shafts. Pass the YZ plane (Fig. 5) through the disc perpendicular to the shaft. When at rest the center of the vertical shaft is at the origin O. When in motion, at a given time the shaft center is at S, with the coördinates (yz); and the center of gravity is at R, a constant distance a from S. Due to the turning of the shaft, the point R revolves around S with the angular velocity p, so that the angle turned through is pt.

14 The force acting on the point S is the spring of the shaft towards the zero position. This is $\frac{W}{\Delta}\overline{OS}$ where $\frac{W}{\Delta}$ is the force of the

shaft per unit of deflection. Since the force from S acts towards the origin, for equilibrium of moments about O the centrifugal force of the disc at R must act in a line through the origin. Also for equilibrium of the two forces they must act in line with each other, be equal in value and be opposed in direction. Then for R to turn about S with the angular velocity p, R must revolve about the origin with the same angular velocity, and in a circle with center at O.

15 The centrifugal force acting at R is $mp^2\ OR$. The component parallel to the Y-axis is $mp^2\ (y + a \sin pt)$; and to the Z axis is $m\,p^2 (z+a \cos pt)$. The spring of the shaft acting at S is $\frac{W}{\Delta}\ OS$, with a component parallel to the Y-axis of $-\frac{W}{\Delta}\ y$; and to the Z-axis, $-\frac{W}{\Delta}\ z$. The sum of these forces along the Y-axis is

$$mp^2\ (y + a \sin pt) - \frac{W}{\Delta}\ y = O$$

and for the Z-axis

$$mp^2\ (z + a \cos pt) - \frac{W}{\Delta}\ z = O$$

Dividing by m and solving for y and z these equations give

$$y = \frac{ap^2}{\frac{g}{\Delta} - p^2} \sin pt \qquad (3)$$

$$z = \frac{ap^2}{\frac{g}{\Delta} - p^2} \cos pt \qquad (4)$$

INTERPRETATION OF EQUATIONS

16 Equations 3 and 4 determine the motion or path of the shaft center. Taken together they are the equations of a circle with center at the origin. Taken separately, the equations are of the form of simple harmonic motion, with a forced vibration of $a \sin pt$ along the Y-axis, and $a \cos pt$ along the Z-axis. $p = \frac{2\pi}{60}$ times the frequency of vibration. The coefficients of the sine and cosine are the amplitude of the vibration along each axis. These are plotted in full lines in

Fig. 3. The amplitude of vibration, being the same for both axes, contains only the independent variable p. The amplitude will increase as the speed or p increases, until the vibration becomes infinite when $\frac{g}{\Delta} - p^2 =$ zero. This is the same critical-speed condition as in the previous solution, Equation 2. Beyond the infinite value the coefficients become negative and decrease, becoming smaller the higher the speed.

17 This can have the physical interpretation that before the critical speed is reached the center of gravity revolves outside of, or in a larger circle than, the mechanical center of the shaft. Beyond the critical-speed point, the center of gravity rotates inside of, or in a smaller circle than, the shaft center.

18 As previously shown, the critical speed occurs when the rotation synchronizes with the natural period of vibration of the loaded shaft. It may be seen from the curve that when the frequency of the forced vibration $\frac{60}{2\pi} p$ is nearly equal to the frequency of the natural vibration $\frac{60}{2\pi}\sqrt{\frac{g}{\Delta}}$ we have a similar state of things to that which gives rise to *resonance* in acoustic instruments and electrical circuits.

19 The natural period of vibration and the forced vibration are the same for either a vertical or a horizontal position of the shaft, so that the same critical-speed formulæ apply for either position. When vertical, the center of the vibration or of the rotation is at $y = 0$, $z = 0$; when horizontal, the center of the vibration along each axis is at $y = -\Delta, z = 0$. The horizontal position is equivalent to a change of coördinate axes from $y = 0$ to $y = -\Delta$, so that Equation 3 becomes

$$y = \frac{ap^2}{\frac{g}{\Delta} - p^2} \sin pt - \Delta \quad \ldots\ldots\ldots\ldots (3a)$$

20 Vibration is caused by an unbalance of the body, or by the center of gravity not coinciding with the mechanical center of the shaft. The centrifugal force of the unbalance causes an accelerating force along each axis, or a forced vibration of a amplitude. This forced vibration causes the shaft to vibrate along each axis with the amplitude of $\frac{ap^2}{\frac{g}{\Delta} - p^2}$. This shaft vibration is in turn the vibration

that is forced on the frame or supporting structure and causes it to vibrate. This latter value is therefore the vibration to be considered in the design of machines. Comparing the two curves of Fig. 3 it will be noted that at zero speed the vibration in Equation 1 is a; in Equation 4 it is zero. Beyond the critical speed, Equation 1 approaches zero; Equation 4 approaches a. This difference is due to Equation 1 considering the motion of the center of gravity, and Equation 4 the motion of the shaft center. Equation 4 is the vibration impressed on the frame and therefore the value to be considered.

TWO CONCENTRATED LOADS

21 Equations covering any two concentrated loads, with either two or three bearing supports, may be developed by the same method as in the previous case. Take the condition shown in Fig. 6, with two discs for concentrated loads and three bearing points. To distinguish

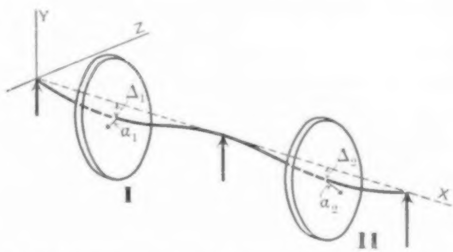


Fig. 6 Two Concentrated Loads on Horizontal Shaft

symbols, let the $Y\ Z$ plane passing through disc I be represented by sub-letters 1, and the plane through disc II by sub-letters 2. To determine the influence one disc has upon the other the equations $F_1 = K_1\ y_1 + K_3\ y_2$ and $F_2 = K_2\ y_2 + K_3\ y_1$ are taken, where y_1 and y_2 are any positions of the shaft deflections on the Y-axis, and F_1 and F_2 are the shaft forces due to the deflections which act toward the unloaded or zero position. K_1, K_2 and K_3 are constants for a given shaft and can be deduced from the deflection equations of beams for different loads and supports. Assume the coördinates of the shaft center in any position y_1, z_1, in the plane through disc I and y_2, z_2 in the plane through disc II. Take the distances from the centers of gravity to the shaft centers to be a_1 and a_2 and their directions to differ by x deg. on the $Y\ Z$ plane. The forces acting on the discs when the shaft is vertical are:

First, centrifugal force:

Plane I, Y-axis, $m_1 p^2 (y_1 + a_1 \sin pt)$;
Z-axis, $m_1 p^2 (z_1 + a_1 \cos pt)$.
Plane II, Y-axis, $m_2 p^2 [y_2 + a_2 \sin (pt + \alpha)]$;
Z-axis, $m_2 p^2 [z_2 + a_2 \cos (pt + \alpha)]$.

Second, reaction or spring of the shaft:

Plane I, Y-axis, $K_1 y_1 + K_3 y_2$; Z-axis, $K_1 z_1 + K_3 z_2$.
Plane II, Y-axis, $K_2 y_2 + K_3 y_1$; Z-axis, $K_2 z_2 + K_3 z_1$.

The summation of these forces along the axes gives the following equations:

$$m_1 p^2 (y_1 + a_1 \sin pt) - K_1 y_1 - K_3 y_2 = O.$$
$$m_1 p^2 (z_1 + a_1 \cos pt) - K_1 z_1 - K_3 z_2 = O.$$
$$m_2 p^2 [y_2 + a_2 \sin (pt + \alpha)] - K_2 y_2 - K_3 y_1 = O.$$
$$m_2 p^2 [z_2 + a_2 \cos (pt + \alpha)] - K_2 z_2 - K_3 z_1 = O.$$

The solution of these equations gives:

$$y_1 = A \sin pt - B \cos pt \quad [5]$$
$$z_1 = B \sin pt + A \cos pt \quad [6]$$
$$y_2 = C \sin pt + D \cos pt \quad [7]$$
$$z_2 = -D \sin pt + C \cos pt \quad [8]$$

where

$$A = \frac{(K_2 - m_2 p^2)\, m_1 a_1 p^2 - K_3 m_2 a_2 p^2 \cos \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$B = \frac{K_3 m_2 a_2 p^2 \sin \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$C = \frac{(K_1 - m_1 p^2)\, m_2 a_2 p^2 \cos \alpha - K_3 m_1 a_1 p^2}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$D = \frac{(K_1 - m_1 p^2)\, m_2 a_2 p^2 \sin \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

INTERPRETATION OF EQUATIONS

22 Equations 5 to 8 are of the form of harmonic motion along their respective axes. Equations 5 and 6 taken together (also Equations 7 and 8) are equations of a circle with center at the origin, the radius of the circle being $\sqrt{A^2 + B^2}$.

23 The coefficients which represent the radii in rotation, or the amplitude of vibration, have the same denominators. The value of the coefficients becomes infinite when the denominators equal zero, which is the critical-speed condition. That is,

$$(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2 = O.$$

$$p = \sqrt{\frac{g}{2}\left[\frac{K_1}{W_1} + \frac{K_2}{W_2} \pm \sqrt{\left(\frac{K_1}{W_1} - \frac{K_2}{W_2}\right)^2 + \frac{2\,K_3^2}{W_1 W_2}}\right]} = \frac{2\pi}{60} N$$

$$N = 132.3 \sqrt{\frac{K_1}{W_1} + \frac{K_2}{W_2} \pm \sqrt{\left(\frac{K_1}{W_1} - \frac{K_2}{W_2}\right)^2 + \frac{2\,K_3^2}{W_1 W_2}}} \qquad \ldots\ldots\ldots [9]$$

for inch, pound, minute units.

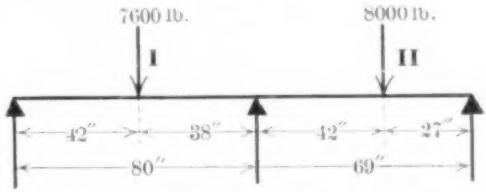


FIG. 7 CONDITION OF LOAD ON SHAFT (FIG. 6)

24 The $\pm$ sign of this equation gives two values of critical speed. This equation is general for two concentrated loads regardless of the method of support, for either two of three bearings.

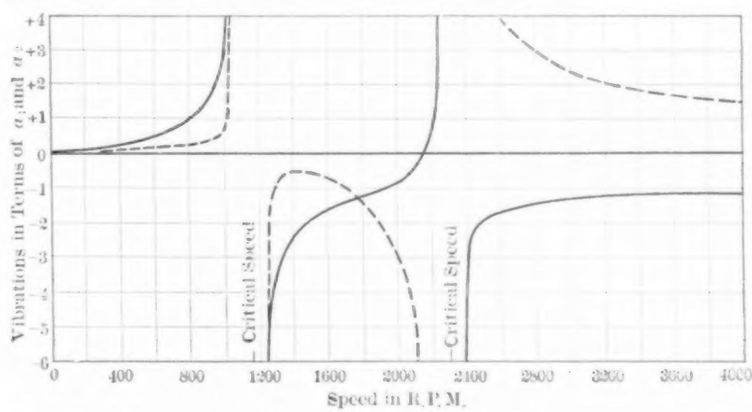


FIG. 8 AMPLITUDE OF VIBRATION WITH TWO CONCENTRATED LOADS (FIGS. 6 AND 7)

SOLID LINE FOR LOAD I; DOTTED LINE FOR LOAD II

25 When the unbalances a_1 and a_2 of the centers of gravity of both loads lie in the same plane, either on the same or opposite sides of the shaft, so that the angle α is zero or π, the coefficient B becomes zero and the radius of the circle of the shaft path is A. This gives below the critical-speed value the smallest circle when the unbalances are on the same side, or α equal to zero degrees; above the critical speed, π gives the smallest circle.

26 The properties of Equations 5 and 8 can be shown more fully by an example. For the conditions given in Fig. 7 the amplitude of the vibrations is plotted in Fig. 8. This machine showed excessive vibration between 1100 and 1200 r.p.m., when not in nearly perfect balance. It could not be speeded up to the second critical speed, the second value being too far above the running speed. The solid curve is the vibration of Load I; the dotted curve is the vibration of Load II. The amount of vibration is in terms of the unbalance a_1 and a_2.

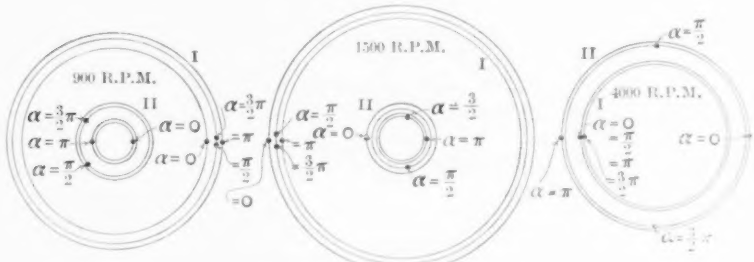


FIG. 9, 10, 11 PATH AND RELATIVE LOCATION OF SHAFT CENTERS FOR DIFFERENT ANGLES OF UNBALANCE

For the same unbalance, Load I, having the weaker portion of the shaft, has the largest vibrations until the first critical speed is reached. Beyond the first critical value the vibrations of Load II become large and influence the vibrations of Load I, reducing them through zero from a negative to a positive value. Beyond the second critical speed, Load II with the stiffer shaft has the larger vibrations.

27 Another interesting thing is the relative location of the shaft centers at any given time. Figs. 9, 10 and 11 show the paths of the shaft centers and their location on the circles, for the angle between the unbalances of $\alpha = 0, \frac{1}{2}\pi, \pi$ and $\frac{3}{2}\pi$, when $pt = 0, 2\pi$, etc. The rotation of the points on all circles is in the same direction as the rotation of the machine. When below the critical speed (Fig. 9),

Load I on the weaker shaft, or the shaft with the largest static deflection, rotates in the larger circle. Note the relative positions of the shaft centers for different values of α: and the influence the larger circles have upon Load II in forcing the unbalances towards opposite sides of the shaft as shown by the positions for $\alpha = \frac{1}{2}\pi$ and $\frac{3}{2}\pi$. Between the two critical speeds (Fig. 10), the positions for all values of α have turned through 180 deg., except Load II, $\alpha = \frac{1}{2}\pi$ and $\frac{3}{2}\pi$. Above the second critical speed the positions of Load II turn through another 180 deg., while Load I is unchanged. Here the stiffer shaft, or Load II, has the larger circles of rotation.

CONSTANTS FOR TWO LOADS

28 As the sixty-nine formulæ given in the table for calculations cannot be derived for want of space, an example will be given to illustrate the method. Take the condition of two loads just considered, using the letters given in Fig. 12 for dimensions and weights. The

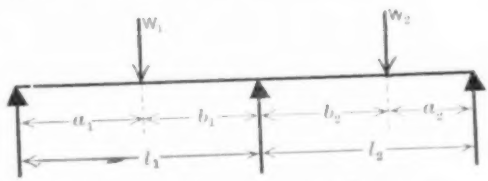


FIG. 12 CONDITIONS OF LOADING

force of the spring of the shaft is $F_1, = K_1\, y_1 + K_3\, y_2$ at W_1, for any deflections y_1 and y_2; and $F_2 = K_2\, y_2 + K_3\, y_1$ at W_2 for any position y_1 and y_2. The standard equations for deflections at the loads are:

$$\Delta_1 = \frac{a_1\, b_1}{6E\, Il_1}\,[2\, a_1\, b_1\, W_1 - m\,(l_1 + a_1\,)]$$

$$\Delta_2 = \frac{a_2\, b_2}{6E\, Il_2}\,[2\, a_2\, b_2\, W_2 - m\,(l_2 + a_2)]$$

$$m = \frac{1}{2\,(l_1 + l_2)}\left[\frac{W_1\, a_1\, b_1}{l_1}\,(l_1 + a_1) + \frac{W_2\, a_2\, b_2}{l_2}\,(l_2 + a_2)\right]$$

Making variable by changing Δ_1 to y_1, Δ_2 to y_2, W_1 to F_1, W_2 to F_2, and solving for F_1 and F_2, gives equations of the above form, where

$$K_1 = \frac{C\, l_1^2}{a_1^2\, b_1^2}[4\; l_2\,(l_1 + l_2) - (l_2 + a_2)^2]$$

$$K_2 = \frac{C\; l_2^2}{a_2^2\, b_2^2}[4\, l_1\,(l_1 + l_2) - (l_1 + a_1)^2]$$

$$K_3 = \frac{C\; l_1\; l_2}{a_1\, b_1\, a_2\, b_2}\,(l_1 + a_1)\,(l_2 + a_2)$$

$$C = \frac{3\, E\, I}{4\; l_1\; l_2\;(l_1 + l_2) - l_1\;(l_2 + a_2)^2 - l_2\;(l_1 + a_1)^2}$$

Constants for other dimensions, loads and supports, may be derived from the deflection formulæ in a similar manner.

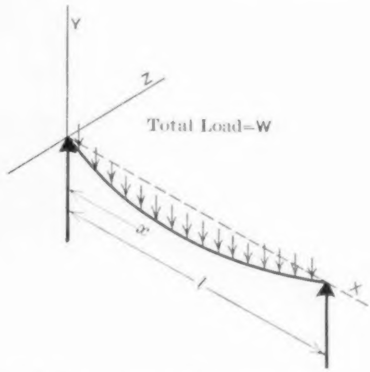


FIG. 13 SHAFT WITH UNIFORM LOAD

DISTRIBUTED LOADS

29 All critical speed formulæ so far developed for distributed loads are based on the equation in Mechanics that

$$E\,I\,\frac{d^4\,y}{d\,x^4} = \text{the weight or force } on \text{ a unit length of beam}$$

and there is an unbalance caused by the center of gravity not coinciding with the shaft center.

30 To simplify the calculations, assume the conditions of Fig. 13, of a shaft with constant diameter uniformly loaded over its entire length by a load, as disc wheels, which will not affect the flexibility of the shaft; and that the centers of gravity of all the discs lie in the same plane and on the same side of the shaft, at a constant distance a

from the shaft center, similar to the single disc in Fig. 4. W = total weight of shaft and discs.

31 Taking any unit length along the X axis, its mass is $\frac{W}{l\,g}$. Assume the center of rotation to be at the origin of the axes when the shaft is vertical. The centrifugal force of the mass of unit length is $\frac{W}{l\,g}\,p^2$ times the radius to center of gravity. This radius projected on the Y axis is $(y + a \sin pt)$ and on the Z axis is $(z + a \cos pt)$, where y and z are coördinates of the shaft center for the shaft in any position of rotation.

32 The forces acting on a unit length, projected on the axes, are:

First, centrifugal force: Y axis, $\frac{W\,p^2}{l\,g}(y + a \sin pt)$; Z axis, $\frac{W\,p^2}{l\,g}$ $(z + a \cos pt)$.

The second force, the spring of the shaft, does not enter as we are considering the forces acting *on* the shaft, and not the reaction of the shaft.

33 The equation in Mechanics of the forces acting on a unit length gives for the Y axis:

$$E\,I\,\frac{d^4y}{dx^4} = \frac{W}{l}\,\frac{p^2}{g}\,(y + a \sin pt)$$

and for the Z axis:

$$E\,I\,\frac{d^4z}{d\,x^4} = \frac{W\,p^2}{l\,g}\,(z + a \cos pt)$$

The general solutions of these equations are:

$$y = [Fe^{kx} + Ge^{-kx} + H \cos kx + J \sin kx - a] \sin pt \ldots\ldots [10]$$

$$z = [Fe^{kx} + Ge^{-kx} + H \cos kx + J \sin kx\text{-}a] \cos pt \ldots\ldots\ldots\ldots [11]$$

where

$$k = \sqrt[4]{\frac{W\,p^2}{E\,I\,g\,l}} \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

e = the base of the natural system of logarithms, and the capital letters are constants determined by the conditions imposed on the equations by the supports, etc., as shown in the following special cases.

SHAFT SUPPORTED AT BOTH ENDS

34 A shaft supported at both ends, as shown in Fig. 13, either vertical or horizontal, imposes the conditions of deflection y, and moment $EI\,\dfrac{d^2\,y}{d\,x^2}$, both equal to zero at the supports, or when x equals zero and when x equals l. This gives four equations with four unknown constants as follows:

$$\text{For } x = 0,\ y = 0$$
$$0 = F + G + H - a$$

$$\text{For } x = 0,\ \frac{d^2\,y}{d\,x^2} = 0$$
$$0 = F + G - H$$

$$\text{For } x = l,\ y = 0$$
$$0 = Fe^{kl} + Ge^{-kl} + H\cos kl + J\sin kl - a$$

$$\text{For } x = l,\ \frac{d^2\,y}{d\,x^2} = 0$$
$$0 = Fe^{kl} + Ge^{-kl} - H\cos kl - J\sin kl$$

The solution of these four equations gives

$$F = -\frac{a}{2}\left(\frac{e^{-kl} - 1}{e^{kl} - e^{-kl}}\right) \qquad G = \frac{a}{2}\left(\frac{e^{kl} - 1}{e^{kl} - e^{-kl}}\right)$$

$$H = \frac{a}{2} \qquad J = \frac{a}{2}\left(\frac{1 + \cos kl}{\sin kl}\right)$$

These values substituted in the general equations 10 and 11 for distributed loads give values for y and z which represent the path of the shaft center for any point x along the length. The coefficients of sin pt and cos pt are the amplitude of vibration along each axis, or the radius of the circle in which any point on the shaft center rotates. These coefficients become infinite or have the critical speed value when $\sin kl = 0$, or whenever $kl = \pi,\ 2\pi,\ 3\pi$, etc. Since p is proportional to k^2 by equation 12 we have an infinite number of critical speeds which have the ratio $1 : 2^2 : 3^2 : 4^2 \ldots$ The first or lowest critical speed is found from

$$k\,l = \pi = \sqrt{\frac{W\,p^2}{E\,I\,g\,l}}\;l$$

For a circular shaft of d in. diameter, $E = 29{,}000{,}000$, $g = 386$,

$$p = \frac{2\pi}{60} N_1$$

$$N_1 = 2{,}232{,}510\, d^2 \sqrt{\frac{1}{W\, l^3}}$$

for inch, pound, minute units. For a shaft with its own weight only, $W = 0.28\, \frac{\pi}{4}\, d^2\, l$. Substituting gives

$$N_1 = 4{,}760{,}000\, \frac{d}{l^2}$$

35 The values of the constants F, G, H and J, inserted in Equation 10, show that the shaft rotates in a bowed condition up to the

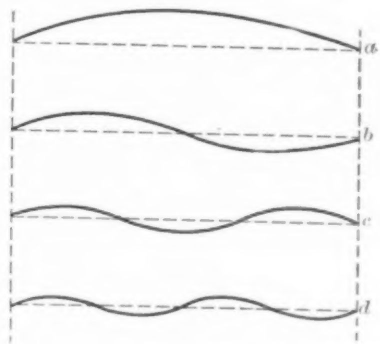


FIG. 14 CONDITION OF ROTATION AT VARIOUS SPEEDS

first critical speed as shown in Fig. 14a; between the first and second critical speeds as in Fig. 14b; between the second and third critical speeds as in Fig. 14c; and so on.

SHAFT FIXED AT BOTH ENDS

36 A shaft with uniform load between long rigid bearings, we can treat as a beam fixed at both ends and impose the conditions upon Equations 10 and 11 of the deflection $y = 0$, when $x = 0$ and when $x = l$; and the tangent to deflection curve $\frac{dy}{dx} = 0$, when $x = 0$ and

when $x = l$. Solving these four equations for the unknown constants, each constant has the denomination of $2 - (e^{kl} + e^{-kl}) \cos kl$. y and z become infinite when the denominator equals zero, or

$$\cos kl = \frac{2}{e^{kl} + e^{-kl}} = \text{sech } kl$$

37 To satisfy this equation kl is nearly $\frac{3}{2}\pi, \frac{5}{2}\pi, \frac{7}{2}\pi, \frac{9}{2}\pi \ldots$ The critical speeds have the ratio:

$$3^2 : 5^2 : 7^2 : 9^2 \quad \ldots . = 1 : 2.78 : 5.45 : 9 \ldots .$$

The first critical speed is when

$$kl = \frac{3}{2}\pi = \sqrt[4]{\frac{W}{E\,I}\frac{p^2}{gl}}\; l$$

$$N_1 = 4{,}979{,}250\, d^2 \sqrt{\frac{1}{W\, l^3}}$$

and for a shaft with its own weight only, $N_1 = 10{,}616{,}740 \dfrac{d}{l^2}$ in inch, pound, minute, units.

OVERHANGING SHAFT FIXED AT ONE END

38 Taking the origin of the coördinate system at the support, we can impose upon Equations 10 and 11 the conditions of a cantilever beam; that is, the deflection $y = 0$ for $x = 0$; tangent to elastic curve $\dfrac{dy}{dx} = 0$ for $x = 0$; the bending moment $\dfrac{d^2 y}{dx^2} = 0$ for $x = l$; and the shear $\dfrac{d^3 y}{dx^3} = 0$ for $x = l$. Solving these four equations for the unknown constants, each constant has the denominator of $2 + (e^{kl} + e^{-kl}) \cos kl$. y and z are infinite when the denominator is zero, or when

$$\cos kl = -\frac{2}{e^{kl} + e^{-kl}} = -\text{sech } kl$$

39 The smallest value of kl to satisfy this equation is 1.8751. The next values are nearly $\frac{3}{2}\pi, \frac{5}{2}\pi, \frac{7}{2}\pi \ldots\ldots$ Critical speeds have

the ratio of 1 : 6.34 : 17.6 : 43.6...... The first critical speed is when

$$kl = 1.8751 = \sqrt{\frac{W p^2}{E\ I\ gl}}\ l$$

$$N_1 = 795{,}196\ d^2 \sqrt{\frac{1}{W\ l^3}}$$

and for a shaft with its own weight only, $N_1 = 1{,}695{,}514\ \frac{d}{l^2}$ in inch, pound, minute, units.

SHAFT FIXED AT ONE END AND SUPPORTED AT THE OTHER

40 With the origin of the coördinate system at the fixed end of the shaft, we can place on Equations 10 and 11 the condition of deflection $y = 0$ for $x = 0$ and for $x = l$; the tangent to the elastic curve $\frac{dy}{dx} = 0$ for $x = 0$; and the moment $\frac{d^2y}{dx^2} = 0$ for $x = l$. Solving these four equations for the unknown constants, each constant has a denominator of $\cosh kl \sin kl - \sinh kl \cos kl$. which equals zero for the critical speeds; or $\tan kl = \tanh kl$

$$kl = \frac{5}{4}\pi, \frac{9}{4}\pi, \frac{13}{4}\pi,$$

The critical speeds have the ratio

$$5^2 : 9^2 : 13^2 : 17^2$$

or

$$1 : 3.24 : 6.8 : 11.6$$

The first critical speed is for

$$kl = \frac{5\pi}{4} = \sqrt[4]{\frac{W\ p^2}{E I g l}}\ l$$

$$N, = 3{,}482{,}715\ d^2 \sqrt{\frac{1}{W\ l^3}}$$

and for shaft with its own weight only

$$N_1 = 7{,}021{,}600\ \frac{d}{l^2}$$

for inch, pound, minute, units.

GENERAL OBSERVATIONS

41 All formulæ developed for critical speed, for both concentrated and distributed loads, apply to vertical shafts as well as horizontal. When the shaft is vertical the equation for the Y-axis only is affected, the value of y dropping the $(-\Delta)$, the coefficient of sin pt being unchanged. Since this coefficient determines the critical speed value, we have the same critical speed for horizontal as for vertical shafts. Although some formulæ use the static deflection Δ, this is an equivalent deflection and can be used for vertical shafts by considering them horizontal.

42 The obliquity of the loads caused by the bending of the shaft has not been considered. When the load is near the bearings, as shown in Fig. 15, the load passes from the full line to the dotted line position, and back to the full line, for each revolution. The inertia of the disc offers a resistance to this change of position; and this resistance raises

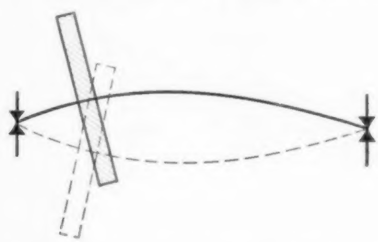


FIG. 15 OBLIQUITY OF LOAD DURING ROTATION

the value of the critical speed. But the obliquity does not introduce a considerable error if the loads are nearly half way between the bearings.

43 Theoretically the vibrations become infinite at the critical speed; actually they do not, but the vibrations are at a maximum point. As shown by the curves of Fig. 3 and Fig. 8, the vibrations will begin at a certain speed, increase as the speed increases, and with still increasing speed will after a while die away. The vibrations may be felt over a considerable range, and the exact point of maximum value is difficult to detect. It is therefore advisable to keep the running speed at least 20 per cent away from the critical value; and if the normal speed is between two critical values, as in Fig. 8, careful calculations should be made for the point of minimum vibration.

44 Under ordinary circumstances the speed should be considerably below the critical value, as then the balance need not be particu-

larly good. When the speed is considerably above the critical value, the vibration is almost proportional to the unbalance (a in the equations) and the flexibility of the shaft; and the balance should be good, to prevent injury to the shaft and excessive vibration when passing through the critical speed.

45 A machine may be run very close to or at the critical speed, but the alignment and play of bearings, all mechanical details and the balance will require extra care, so that a troublesome and more expensive machine results before it is in good operating condition. The machine will run smoothly for a considerable time, until some mechanical fit or play cause a slight unbalance and immediately sets up excessive vibrations.

46 All of the solutions of shaft deflection in this paper are in the mathematical form of a harmonic vibration produced by the impressed vibration of the unbalance of load. Harmonic vibrations of this form have a special solution by calculus when p^2 equals the natural period of vibration of the shaft, or in all the cases considered, the critical-speed period. For Equations 3 and 4, when $\frac{g}{\varDelta} = p^2$, the special solutions are

$$y = \frac{a\,t}{2\sqrt{\frac{g}{\varDelta}}} \sin \sqrt{\frac{g}{\varDelta}}\,t$$

$$z = \frac{a\,t}{2\sqrt{\frac{g}{\varDelta}}} \cos \sqrt{\frac{g}{\varDelta}}\,t$$

These equations show that during the critical-speed period the vibrations increase theoretically with the time, so that in machines running above the critical speed there is less vibration at the critical-speed point when it is rapidly passed over. The equations also show a transfer of energy; the kinetic energy from the unbalance being transformed into the potential energy of the shaft deflection, so that a machine with *nearly perfect* balance may run smoothly for considerable time at the critical speed before vibrations appear. The writer has not seen or had sufficient proof of the action of these two equations, but they may explain some of the peculiar phenomena observed in the vibration of certain machines.

47 With excessive vibration in passing through the critical speed

there is a considerable tendency to spring the shaft by giving it a permanent set. This is most dangerous when the machine is first started, before it has a running balance. Partly for this reason many designers use the more expensive nickel-steel forged shaft instead of carbon steel. With due consideration of the smaller coefficient of expansion of nickel-steel, in distorting large shafts when all parts are not at the same temperature, and of the fatigue or reversal of fibre stress in horizontal shafts, the machine and shaft can be so proportioned for smooth running that the finer grade of shaft steel is not always necessary.

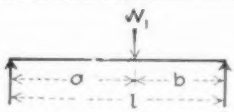
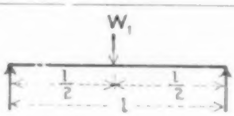
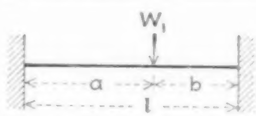
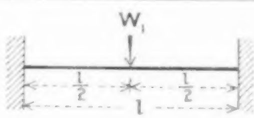
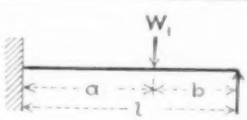
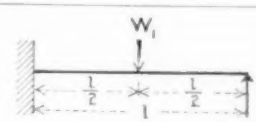
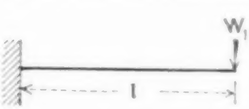
TABLE OF FORMULÆ

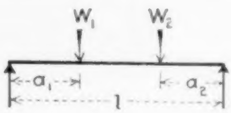
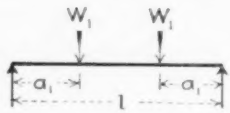
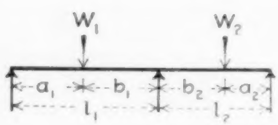
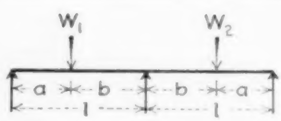
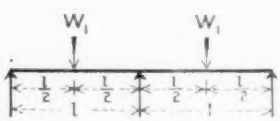
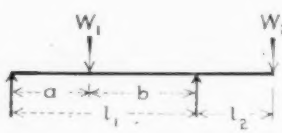
48 The formulæ developed in this paper have been transformed to suit a number of special conditions, and placed in tabulated form for convenient use. The data required for the solution of critical-speed problems are the same as those for shaft deflection at loads. As the shaft is usually of variable diameter, and its stiffness is increased by a long hub, an ideal shaft of uniform diameter and equal stiffness, or for the same deflection, must be assumed. The loads are usually concentrated with an ideal point of application. The weights and distances between bearings and loads are the same in the ideal as in the actual case. Experience has shown that when the largest shaft diameter and uniform load cover about one-third of the span, approximately the same deflection is given for the load concentrated with a uniform shaft of the largest diameter. The weight of the shaft can be divided among the concentrated loads. As formulæ have not been developed for more than two loads, when more than two loads are given they must be transformed into two resultant loads that would give the same deflection. For this case, two critical speeds are found, one of which is usually far from the working speed.

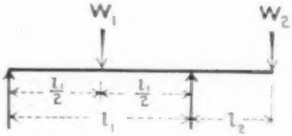
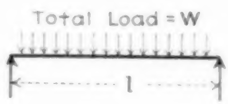
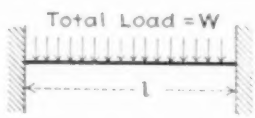
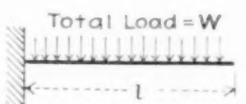
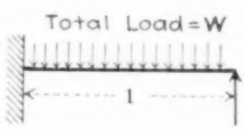
CRITICAL SPEED FORMULAE

Weights in Pounds, Dimensions in Inches, Vertical Shafts Considered Horizontal

N, N_1, N_2 = critical speeds in r.p.m.
Δ_1, Δ_2 = static deflections at W_1 and W_2 (shaft horizontal).
d = diameter of shaft (inches). $E = 29{,}000{,}000$.

Single Concentrated Load General Formulae	$N_1 = \frac{187.7}{\sqrt{\Delta_1}}$	1
	$N_1 = 387{,}000\frac{d^2}{ab}\sqrt{\frac{l}{W_1}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 a^2 b^2}{3EIl}$	2 1 3
	$N_1 = 1{,}550{,}500\, d^2\sqrt{\frac{1}{W_1 l^3}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 l^3}{48EI}$	4 1 5
	$N_1 = 387{,}000\, d^2\sqrt{\frac{l^3}{W_1 a^3 b^3}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 a^3 b^3}{3EIl^3}$	6 1 7
	$N_1 = 3{,}100{,}850\, d^2\sqrt{\frac{1}{W_1 l^3}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 l^3}{192EI}$	8 1 9
	$N_1 = 775{,}200\frac{d^2 l}{ab}\sqrt{\frac{l}{W_1 a(3l+b)}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 a^3 b^3}{12EIl^3}(3l+b)$	10 1 11
	$N_1 = 2{,}337{,}000\frac{d^2}{l}\sqrt{\frac{1}{W_1 l}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{7}{768}\frac{W_1 l^3}{EI}$	12 1 13
	$N_1 = 387{,}000\, d^2\sqrt{\frac{1}{W_1 l^3}}$ $N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$ $\Delta_1 = \frac{W_1 l^3}{3EI}$	14 1 15

Two Concentrated Loads General Formulae	N_1 and $N_2 = 132.3\sqrt{\left(\frac{K_1}{W_1}+\frac{K_2}{W_2}\right)\pm\sqrt{\left(\frac{K_1}{W_1}-\frac{K_2}{W_2}\right)^2+\frac{4K_3^2}{W_1W_2}}}$	16
	$C=\frac{6EI}{(l-a_1-a_2)^2[l(3l-2a_1-2a_2)-(a_2-a_1)^2]}$	17
	$K_1=C\frac{2}{a_1^2}l(l-a_2)^2$	18
	$K_2=C\frac{2}{a_2^2}l(l-a_1)^2$	19
	$K_3=C\frac{l(l^2-a_2^2-a_1^2)-a_1a_2(a_1-a_2)}{a_1a_2}$	20
	N_1 and N_2 = Substitute in Equation 16	
	$N_1=548{,}400\frac{d^2}{a_1(l-2a_1)}\sqrt{\frac{I}{W_1}}$	22
	$N_2=548{,}400\frac{d^2}{a_1}\sqrt{\frac{I}{W_1(3l-4a_1)}}$	23
	$N_2=187.7\sqrt{\frac{1}{\Delta_1}}$	24
	$\Delta_1=\frac{W_1a_1^2}{6EI}(3l-4a_1)$	25
	$C=\frac{3EI}{4l_1l_2(l_1+l_2)-l_1(l_2+a_2)^2-l_2(l_1+a_1)^2}$	26
	$K_1=\left(\frac{l_1}{a_1b_1}\right)^2C[4l_2(l_1+l_2)-(l_2+a_2)^2]$	27
	$K_2=\left(\frac{l_2}{a_2b_2}\right)^2C[4l_1(l_1+l_2)-(l_1+a_1)^2]$	28
	$K_3=\frac{l_1l_2}{a_1b_1a_2b_2}C(l_1+a_1)(l_2+a_1)$	29
	N_1 and N_2 = Substitute in Equation 16	
	$C=\frac{3EIl}{2a^2b^2[4l^2-(l+a)^2]}$	30
	$K_1=K_2=C[8l^2-(l+a)^2]$	31
	$K_3=C(l+a)^2$	32
	N_1 and N_2 = Substitute in Equation 16	
	$N_1=1{,}547{,}000\,d^2\sqrt{\frac{I}{W_1l^3}}$	33
	$N_1=124.3\sqrt{\frac{1}{\Delta_1}}$	34
	$N_2=2{,}337{,}000\,d^2\sqrt{\frac{I}{W_1l^3}}$	35
	$N_2=187.7\sqrt{\frac{1}{\Delta_1}}$	36
	$\Delta_1=\frac{7}{768}\frac{Wl^3}{EI}$	37
	$C=\frac{12EI}{a^2l_2^2[4b^2l_1(l_1+l_2)-(l_1^2-a^2)^2]}$	38
	$K_1=Cl_1^2l_2^2(l_1+l_2)$	39
	$K_2=Ca^2b^2l_1$	40
	$K_3=\frac{C}{2}al_1l_2(l_1^2-a^2)$	41
	N_1 and N_2 = Substitute in Equation 16	
	$\Delta_1=\frac{W_1a^2b^2}{3EIl_1}-W_2\frac{al_2}{6EIl_1}(l_1^2-a^2)$	42
	$\Delta_2=\frac{W_2l_2^2}{3EI}(l_1+l_2)-W_1\frac{a\,l_2}{6EIl_1}(l_1^2-a^2)$	43

	$C = \dfrac{3EI}{l_1^3 l_2^2 (l_1+l_2) - \frac{9}{16} l_1^4 l_2^2}$	44
	$K_1 = 16 C l_2^2 (l_1 + l_2)$	45
	$K_2 = C l_1^3$	46
	$K_3 = 3 C l_1^2 l_2$	47
	N_1 and N_2 = Substitute in Equation 16	
	$\Delta_1 = \dfrac{W_1 l_1^3}{48EI} - \dfrac{W_2 l_2 l_1^2}{16EI}$	48
	$\Delta_2 = \dfrac{W_2 l_2 (l_1+l_2)}{3EI} - \dfrac{W_1 l_2 l_1^2}{16EI}$	49
Distributed Loads.— Δ = *Maximum Static Deflection.*		
	$N_1 = 2{,}232{,}510\, d^2 \sqrt{\frac{l}{Wl^3}}$	50
	$N_1 = 211.4 \sqrt{\frac{1}{\Delta}}$	51
	$N_1 = 4{,}760{,}000 \frac{d}{l^2}$ (Shaft alone)	52
	$N = [1, 4, 9, 16, \text{Atc.}] N_1$	53
	$\Delta = \frac{5}{384} \frac{Wl^3}{EI}$	54
	$N_1 = 4{,}979{,}250\, d^2 \sqrt{\frac{I}{Wl^3}}$	55
	$N_1 = 245 \sqrt{\frac{1}{\Delta}}$	56
	$N_1 = 10{,}616{,}740 \frac{d}{l^2}$ (Shaft alone)	57
	$N = [1, 2.78, 5.45, 9. \text{Atc.}] N_1$	58
	$\Delta = \frac{Wl^3}{384EI}$	59
	$N_1 = 795{,}196\, d^2 \sqrt{\frac{I}{Wl^3}}$	60
	$N_1 = 167.6 \sqrt{\frac{1}{\Delta}}$	61
	$N_1 = 1{,}695{,}514 \frac{d}{l^2}$ (Shaft alone)	62
	$N = [1, 6.34, 17.6, 43.6, \text{Atc.}] N_1$	63
	$\Delta = \frac{Wl^3}{8EI}$	64
	$N_1 = 3{,}482{,}715\, d^2 \sqrt{\frac{I}{Wl^3}}$	65
	$N_1 = 209.7 \sqrt{\frac{1}{\Delta}}$	66
	$N_1 = 7{,}021{,}600 \frac{d}{l^2}$ (Shaft alone)	67
	$N = [1, 3.24, 6.8, 11.6, \text{Atc.}] N_1$	68
	$\Delta = \frac{Wl^3}{185EI}$	69

THE TRAINING OF MEN—A NECESSARY PART OF THE MODERN FACTORY SYSTEM

By Magnus W. Alexander, Published in The Journal for January 1910

ABSTRACT OF PAPER

The paper outlines briefly the educational policy of the General Electric Company at Lynn, Mass., where a systematic training is provided, suitable to all classes of people. The unskilled worker without particular education receives a training adequate to his immediate needs; the grammar school boy is initiated into the trades on the basis of a four years' course with educational instruction of a high school character; the high school graduate is trained for semi-professional service of a technical or business nature, on the basis of a three years' course with educational instruction of collegiate grade; and the college graduate is prepared for professional service of the highest order, on the basis of a two years' training of the character of a post-graduate course. An hour and a half to two hours must be spent in the classrooms every day, except Saturday, and except during parts of July and August, and classes meet during regular working hours, the students receiving the same compensation as during working hours. The apprentice training room is a trade school in the factory and permits of training under the most favorable conditions and expert supervision.

DISCUSSION

Prof. Ira N. Hollis said that this movement, which practically began ten or fifteen years ago, is bound to extend throughout the country. He referred to the city of Geneva, one of the best governed in the world, where the city council directs these courses of training, and said that the future would show whether our cities could undertake this. In the meantime these great companies are stepping into the gap.

2 The college graduate, as Mr. Alexander has pointed out, lacks practical knowledge. A committee appointed some years ago to represent all the engineering societies of England, recommended that every engineer spend at least one year in a commercial establishment before graduation, this year to follow the first year of

This paper was presented at Boston, March 11, 1910. The discussion is given in abstract only.

college work. Although this seems a prolongation, it is rather a shortening of the course, because of the value of the commercial contact. It might be difficult to get the larger firms to take men under such conditions, but it is probable that such firms would realize that they would profit very largely by the return of men thus trained into business, through their knowledge of the firm's products. Every industry and every side of our industrial life ought to have a school in an establishment where the product is made, not only for the workmen but for the graduates who are going to become engineers in that specialty.

H. S. Knowlton.[1] The industrial organization needs educational advantages among its workers. The public service corporation is coming to the same position, and within the last two or three years quite a number of companies working along street railway lines, or in central station operation, have taken this question up with considerable interest.

2 To speak specifically, perhaps one of the most interesting examples in this vicinity is afforded by the Boston Elevated Railway Company. About four years ago the company began a series of car-house foreman meetings, held monthly under the supervision of John Lindel, the superintendent of rolling stock and shops. At these meetings the defects of the cars are discussed, diagrams and charts are put on the blackboard, and the improvements in keeping rolling stock free from troubles from one year's end to another shown graphically before the men. After full discussion, the meeting is usually turned over to some representative of the operating department especially familiar with one line or another of the work, who either reads a paper or gives an informal address on some subject of special interest. Of the subjects discussed, three or four that may be cited are snow-plow maintenance, accidents, fire protection, gears and motor troubles.

3 The frank and free discussion of these topics in an intensely active operating company is bound to increase the efficiency of minor subordinate officials. Last October the Boston Elevated Railway Company started another series of meetings for its pit-men and car-house foremen and some selected shop employees. The object of the meetings held weekly has been to increase the theoretical knowledge of the work, in the belief that some classes of employees need

[1] Technical Journalist.

this, while the college graduate perhaps needs an increase in practical knowledge. Subjects that have been taken up in these lectures are static electricity, magnetism, measurement of currents, induction currents, electrical considerations in the design of direct-current motors, rheostatic control, multiple-unit control, air-brake equipment, control equipment, track maintenance, blue-print reading-records to maintain personal efficiency, and the design of various electric motors used on the system for car propulsion. Furthermore, in a shop where the work is so specialized that the foreman can follow the work of each man when piece work is undertaken, he can place the man on his own feet and assist him to make the most of his time and to cut out lost movements and other waste in the handling of his tools.

Henry Eckford Rhodes[1] spoke on the general subject of training men for factory work and referred to the methods of early apprenticeship days and his own hardships in securing a mechanical education. He pointed out the greater opportunities afforded at that time because of the greater thoroughness of the boy's training, a thoroughness which the present methods of specialization render impossible. He thought this subject to be of the greatest importance, since lack of efficient men is blocking the growth of American industries, while the non-trade branches including the professions and business occupations, are over-crowded, a condition due to the restrictions which prevent boys from securing apprenticeships in trades. He advocated the establishment of free or partly free industrial schools such as are being successfully operated in Europe, which tend to supplement the education of an ordinary school with training calculated to make the boy a more useful member of society and a larger contributor to the nation's wealth. The colleges and universities and the leading high schools in cities should be equipped with complete mechanical plants and able instructors and the electives should be extended so that greater opportunities might be given for learning trades. The work at the Naval Academy and in our technical colleges proves that it is not impossible to acquire an education and a trade at the same time and he believed that as a result of such a method of procedure many would take up a trade who are now prevented from doing so by present conditions.

[1] Passed Assistant Engineer, U.S.N., Ret.

Prof. Charles F. Park[1] spoke of the work of the Lowell Institute for Industrial Foremen, an evening school maintained by President Lowell under the auspices of the Massachusetts Institute of Technology, described in the discussion of a paper on College and Apprentice Training by J. P. Jackson (Transactions, Vol. 29, p. 507). This school for foremen was not planned for the great mass of so-called working people, but for the minority who are not uneducated, but who have been unable to gain a technical education. It comprises two courses, mechanical and electrical, each extending over two years, and including lectures, recitations, drawing-room exercises and laboratory practice. The courses are conducted by members of the faculty of the Massachusetts Institute of Technology, which has also given the use of its laboratories and class rooms. The students are required to spend two hours at the school three or four evenings a week and as many more hours in home study. There have been about ninety in the first-year class and sixty in the second class.

2 One hundred and eighty-three men have graduated. That the school is making the men more efficient in their regular occupations, and qualifying them for advancement along the lines in which they are working, has been demonstrated by these graduates. This is a strong endorsement of Mr. Alexander's statement that "training will increase their economic value and contentment, and add materially to the productive efficiency of the factory."

3 The paper presents a scheme for training men which in many ways seems ideal. To what extent this may be practically realized under our factory conditions is, in the writer's opinion, open to some question; for industrial activity, competition and numerous manufacturing considerations, the governing conditions for any plan of shop training, would necessarily influence the educational value of the work. The interest of the superintendent and foremen is in the efficiency and output of their departments; questions of instruction are secondary, and under the present scarcity of capable men, it would probably be difficult to find good foremen who would be good teachers. The students themselves, as the author says, must "never forget that they must earn as well as learn in the service of their employer." These conditions seem to be confronting difficulties in the way of thorough training from the educational standpoint.

4 This is not a criticism of shop courses as a whole, for the writer

[1] Director, Lowell School for Industrial Foremen.

believes certain parts of the man's training can be gained as well, if not better, in the factory than in the college or regular school. No course of study can possibly produce a mechanic and engineer, or an electrician, so well fitted that there is nothing left for him to gain from practical experience. His grasp of practical detail and well-balanced judgment will come only after years of service in the factory. But general training in fundamental principles of mathematics and pure applied sciences can best be given in schools which have this training for their aim and not as a secondary purpose. The mechanical industries should not be called upon to give training beyond that directly associated with their processes and methods. These industries are not philanthropic institutions and they cannot be expected to furnish such general training to our young men as we should demand from our public schools and colleges.

George Clinton Ewing[1] said the Westinghouse Electric and Manufacturing Company had a course very similar to that in the General Electric Company described by Mr. Alexander. Delegates were being sent to the various colleges, outlining the company's plan and asking the students to come to them. There are lectures in the evening rather than the day time with various topics for discussion. The students publish the Electric Club Journal, which costs more than is realized from its sale, the deficit being made up by the company. The engineers in the factory give talks and provide papers for the meetings.

Prof. E. F. Miller said that some of the Westinghouse delegates had come to the Massachusetts Institute of Technology and the prospect offered the students seemed a good one. By the company's new arrangement the men are shifted every two months, so that in the course of a year or so they have spent practically two months in every department. The students are to be put in charge of the small testing units up to 500 kw., with a more experienced man over them, and will be called upon to fill vacancies higher up in the company, as soon as these occur.

R. H. Smith.[2] The claims of the scheme outlined by Mr. Alexander are broad, and if true will cause a complete reversal of present

[1] Westinghouse Electric & Mfg. Co., Boston.
[2] Massachusetts Institute of Technology.

methods of education. If mechanical trades and technical subjects such as drawing, mechanism, machine design, pattern-making, machine construction, foundry work, steam, electrical and mechanical engineering, can be taught effectually and rapidly in the shop and factory, then schools and colleges are unnecessary and a needless expense, and our leaders in education have been near-sighted and unwise.

2 For hundreds of years the shop, factory and office provided the only opportunities for a boy to learn a trade or acquire a profession. But under the apprenticeship system the process of learning a trade was slow, illogical, and unsatisfactory to both boy and employer. There was no course, instruction, method or system, to the boy's advancement. The amount of skill and knowledge the boy acquired during his apprenticeship depended upon his own thought and calculation and upon accident, as the teaching results of the shop are small. It was pre-supposed that he would acquire information and skill by observing what went on about him, and his attempts to learn were ventures often disastrous and discouraging.

3 The factory system, with its division of labor, its countless duplication of like parts by means of highly perfected special machinery, where the operations are largely repetitional, was the means of the gradual extinction of the old apprenticeship system, and of the supplanting of the long-trained skilful mechanic by the green hand who can in one or two months be broken in to become an operator or machine specialist, and by repetition becomes expert, but with any change of work or machine is helpless.

4 How does the new apprenticeship system differ from the old? In the old system the boy was under the direction of the foreman, assisted by the journeyman of the department; in the new system he is under the direction of an instructor, assisted by an apprentice. In the old system the boy had quite a variety of work selected from a small shop. In the new system the work is selected from a large manufacturing plant and must be to a great extent work of duplition and repetition, a dulling influence on the boy's mind at that formative age when it is imperative to avoid such influences. Neither the old nor the new systems offer any organized course or systematic instruction. The boy in either case, in large measure, must rediscover elementary facts, as would not be permitted in branches of education other than in the mechanical trades. Under the new system the boy gets a few hours a week of book-learning, but pays dearly for it by having his course unnecessarily lengthened and by

doing more repetitional work, as no company could afford to give him this instruction were it otherwise.

5 Because our forefathers came to the conclusion that learning a trade was slow, illogical, unsystematic, and wasteful of time and money, schools were established and courses organized to supply the deficiencies of shop and factory, and very few would think of returning to the old system.

6 In the evolution of the many systems of teaching, the laboratory or problem method has been the most successful. This is the method in use in our engineering and medical schools, our colleges and universities, and no system in the world has trained so many men so well. It has been employed for more than thirty years at the Massachusetts Institute of Technology. A method is worked out by graded lessons from the simple to the complex, so that with the least expenditure of time a sound, systematic acquirement of information and skill is provided. In the lecture room the instructor, pre-supposing the ignorance of a class of students, begins at the foundation, thus developing the mechanical judgment of the students by making, in advance of practice, a careful study of the problem and its solution.

7 In the laboratories all students have equal opportunities to practice and apply the instruction, not as a venture, but with a clear knowledge of the method of solving, obtaining results many times more effective than can be taught in the shop or factory through constructive and repetitional work. The laboratory method creates enthusiasm and class spirit, tremendous factors in acquiring information and skill, and trains students to think logically and plan intelligently.

8 Mr. Alexander's paper says: "The very fact of this work being a part of the commercial output of the factory automatically insures a high standard of quality and quantity, and eliminates the false notions of these values usually found in purely educational trade schools." When I first came from manufacturing into the profession of teaching, I had this same idea. I thought that schools teaching the mechanical and industrial arts created false notions of accuracy, quality and quantity in the minds of the students. I thought the apprenticeship system of the firm employing thousands of men with which I was formerly connected, gave their apprentices training that could not be improved. Time and experience, however, have proved that it was I who had the false notions and ideas, and that the laboratory or problem method for teaching the principles

of the mechanical and industrial arts as a part of an engineering training, or the essentials of a trade, can never be equalled for rapidity by the apprenticeship system, however perfect, or well or generously managed. The shop and factory have not the facilities for a graded course, nor the necessary teaching knowledge. They cannot properly teach beginnings, while the school cannot teach experience. Each has a field of its own. No word is more misleading in industrial education than the word practical. The laboratory or problem method is the most practical kind of work for the learner and the only true solution of the problem of teaching the mechanical trade rapidly, economically and progressively.

9 This apprenticeship system is undoubtedly well managed and produces as good results as may be obtained or expected from any large shop or modern factory system. Mr. Alexander says: "No course has been laid out for practical work, each apprentice being advanced as fast as is consistent with his individual capacity." While there is no course and, consequently, no systematic instruction, still this system has a field of its own in training its own operators and machine specialists. The essentials of a trade may be taught also by this system provided the apprenticeship course is long enough, say, five to seven years. This will enable the boy to obtain sufficient information and skill to meet changing industrial conditions.

10 While this apprenticeship system will undoubtedly attract many boys who have neither the opportunity nor the means to learn a trade by more rapid methods, it will not find a permanent place in a modern system of education. The American people are striving for the ideal in the field of industrial education and many schemes and plans are being tried out with varying success, and I predict that when this industrial education question is settled it will be along the lines of laboratory or problem methods. This whole subject of industrial education is largely one of the repetition of the factory, and the variety of the school. Repetition is the death of ambition, advancement and development; variety is the life of energy, enthusiasm and progress.

Dickerson G. Baker. Many students of industrial conditions believe that the producing capacity of this country may be limited shortly by the number of men available for properly supervising work. This shortage of competent men is far more noticeable and serious among foremen than among higher executives, and, though rarely admitted, the efficiency of a manufacturin6 plant depends

as much upon a high average ability of foremen and gang bosses as upon executive capacity.

2 In the plant with which I am connected, there have been for years apprentice courses, averaging three years in length, for pattern makers, molders and machinists, and we have a one year course in commercial and designing engineering open to technical graduates only. But we have recently instituted and are giving our principal attention to a class of young men who have been carefully selected as having the proper qualifications to become gang bosses, rate-setters, equipment designers and foremen. Candidates must have had two or more years' machine shop practice and have given evidence of energy, thoroughness and a capacity for analysis.

3 Each student, in this course is first assigned to some machine tool, which he is taught to operate properly, and is given written instructions and a form upon which he records detailed time studies of his work, made along lines developed by Fred. W. Taylor, Past-President of the Society. The importance of realizing the full efficiency of the tool upon the class of work being done is impressed upon the student. Records of time taken for the various steps in the operation being performed, are examined daily by the foreman in charge of the class, and opportunities for improving work and saving time are pointed out. In this way the young man rapidly acquires an appreciation of the value of seconds and of the close study of details.

4 As a student becomes proficient with the operation of one tool he is assigned to others under the same conditions. After a reasonable length of time, he is permitted to make time studies of the work of other operators, and from the records of these studies we are able to make improvements in methods and equipment for the operations observed, and to establish fair and equitable piece-rates, not only reducing our manufacturing cost, but enabling our workmen to earn a considerable increase over their normal day rate. We insist upon fair and straightforward methods in dealing with the workmen in regard to these rates, and there has been practically no trouble in applying them as the student is nearly always able to demonstrate personally that the work can be done in the time assigned.

5 It is noticed that foremen trained in this way tend to proceed in an orderly manner in laying out new work in their departments, When informed as to the number of given parts to be made, they are able to determine the relation which should exist between the amount to be spent for special equipment and the amount to be spent for direct labor on the job.

6 The desirability of the student's having one year of practical experience under actual commercial conditions, between the first and second years of his technical course, brought out by Professor Hollis, is certainly one of the greatest truths before us. I should say further that if we could alternate one year of technical instruction with one year of practical experience, the results would be even better, since repetitive experience has certain great advantages absolutely necessary to the perfection of the competent executive of a manufacturing organization.

Prof. Gardner C. Anthony. I believe the plan of the apprenticeship system to be an excellent one, and a step in the right direction. My criticism relates to the amount and character of the theory to be taught in the proposed course. Problems will probably arise similar to those with which the engineering schools have been struggling in adapting the several courses in shopwork to the curriculum, and we find today a great variety of methods employed for giving this instruction.

2 All classes of shop work should be taught for their educational value, rather than for the information which may be acquired through them. If a course in machine tools for example, is not closely articulated with such subjects as physics, mechanics, mechanism, design, etc., and subjected to the same tests for educational development that are given in other courses, this work had better be relegated to the factory. But because such courses can be made of great pedagogical value by properly articulating them with other courses, they have a proper place in the curriculum of the engineering school.

3 The new form of education under discussion may find a like difficulty in incorporating such subjects as analytical and descriptive geometry, elementary calculus, thermodynamics, etc., into the new curriculum of the factory. This is more likely to occur if an attempt is made to duplicate the courses now given in the colleges, using similar text books and conducting the work in the same manner.

4 A course in descriptive geometry might be given which would not occupy more than one-half the time devoted to that subject in engineering schools; and it should be of a different character more closely allied to the problems to be met in the shop, which necessitates a special text book prepared by those in charge of such a course. This same suggestion would apply to analytical geometry. It will require considerable skill on the part of the instructor to make the element of calculus a live topic in the midst of the pressure of the more practical subjects, although I believe that it can be done.

5 If the subjects can be thus closely related, I believe that the new form of training will be well adapted to a large number of young men who will become capable of filling the better class of positions in engineering establishments.

Prof. Peter Schwamb thought the separation of the instruction department from the shop a wise one, since it brings the student under better supervision and his advance can be made more rapid, if not too much repetitional work is required.

2 While the treatment of the students as individuals in their manual work may advance the better ones more rapidly, the usual effect will be that they obtain more of the instructor's time than is their due, since he will naturally want to instruct where his efforts will make the best showing. The adoption of the class method of instruction will beget a friendly spirit of rivalry among the students and save much valuable time of both instructors and students.

3 The advance of the class should be as rapid as is consistent with the production of good work and the thorough mastering of the operation, the element of expertness being left for future training. The student may be made familiar with the fundamentals, his class-room training being properly correlated to his shop work. After such fundamental training the student may be safely started upon commercial work, with a view of obtaining expertness on the various tools.

4 Mr. Alexander's plan can probably be improved by the adoption of definite courses systematically arranged for the beginners in all departments, such courses covering the fundamental principles and operations in logical order, and it is suggested that such a plan be introduced in the foundry instruction department yet to be equipped. Such courses would also offer a greater attraction to technical graduates, who could obtain through them a much desired practical experience and be brought into close touch with commercial engineering work.

Luther D. Burlingame said there is no substitute for patient and continued practice at a trade and that the apprenticeship system is the most satisfactory means of producing skilled workmen. Its efficiency is greatly increased however by combining with it auxiliary training to supplement the main line of work, such as mathematics, drafting, mechanics, etc., for the machinist apprentice, and varied machine shop work for the draftsman apprentice, these illustrations being typical of the needs in all trades.

2 The value of the shop school is that it makes auxiliary training a necessary part of the course of apprenticeship for which the apprentice is paid and during which he is under full control of his employers. This can be acquired in an evening school, but unless compulsory a large number of those needing this training most would not avail themselves of it.

3 There is a wide field for the employment of different methods in carrying out a system of industrial education in the shop. In the Brown & Sharpe works there are about 150 apprentices coming under some form of auxiliary training. The machinists' apprentices, instead of working part of their time in a machine shop apprentice training room, spend all of their four years of apprenticeship, except for school work, in the regular shop departments. This plan does not require the duplication of machines in another department and it keeps the boys in touch with the more advanced machine operations, so that they can gain experience constantly by observing work going on about them as well as from what they are doing themselves. It brings about contact with the various foremen during the entire period and this mutual acquaintance is helpful in determining the boy's value and, from the apprentice's standpoint, in getting into the spirit of the shop and its personnel. During the time these apprentices are working in the machine shop they are given work in eight or more of the regular departments, thus becoming acquainted with foremen and workmen as well as with methods. No shop should be deterred from establishing an apprenticeship system because it does not feel justified in equipping an apprentice training department where a separate equipment of machines is required.

4 In the Brown & Sharpe works, without the use of text books and without the learning of rules, etc., the problems are presented as they would arise in the shop, except that they are in regular sequence as to subject and difficulty. They are taken up with such reference tables and books at hand as should be in the possession of intelligent mechanics, and the boys are taught how to use such means to solve the problems. They are not taught geometry, algebra, trigonometry, etc., as such, but learn quickly to apply such principles of these sciences as are needed for the problems arising in the shop, and perhaps before knowing these sciences even by name are making practical use of them in their work. Instead of learning certain rules and then applying them, the application comes first showing what the rule must be, or, at least, where in the ordinary

reference books it can be found. Bettei still, the method of solving a problem is often worked out by the boys based on what they have previously done, and making them to that extent independent both of memorized rules and reference to text books. The whole course is directed toward cultivating the reasoning powers rather than the memory, and gives a chance for the intelligent grammar school graduate to hold his own better than would be expected, in comparison with the ordinary high-school graduate of the same age. The aim is to make skilled machinists, and while this course fits also for foremanships and other lines of advancement, the greatest need of today is for skilled workmen.

5 The speaker also exhibited several blue-prints showing problems to be worked out by the students. These related to linear measurements, fractions and decimal equivalents, screw threads, tapers, gearing, etc., certain data being given from which discussions or other results are to be determined by calculation.

The Author.[1] It was never my intention nor anybody's intention that technological or other schools should be abolished. Let me refer to one or two things. Mr. Smith claims that the repetitive character of our work, so necessary, is absolutely useless in any attempt at training efficient mechanics in these shops. But does he forget that the very same jigs and fixtures that are needed in such large numbers and high efficiency must be made by intelligent mechanics? It does not matter how much repetition work is produced by these jigs. We must have men who can design these jigs and fixtures and can design proper machinery on which these jigs and fixtures can be used.

2 Also a certain amount of repetitive work is not only not bad, but is absolutely necessary for the proper training of a mechanic. The intensity of production which can be taught only through repetitional work must be applied to our growing generation. It is absolutely necessary. You can't teach the intensity of production and all that it means by having a young man make a special tool or a jig or a fixture, because you have no proper, definite measure of the time it should take. But if you let that young man make 50 motor-shafts, you can hold him down not only to absolute commercial accuracy, but to a very fair degree of speed with his own hands and with his machine. How about the many high school graduates who

[1] This discussion was not revised by the author.

cannot or do not want to go to a technological school or to a college? And how about the 75 per cent of the grammar school graduates who never go beyond the public grammar school because they cannot or do not want to enter our industries? All these people, both the high school and the grammar school boys, making almost 100 per cent, must receive mechanical training, if they receive it at all, in the shop, which is the proper place for them to receive it.

3 Should we leave out the class room work because we cannot give it as well as it is given in college? And, let us say, we do not give it as well, because we are not professional teachers. We are professional business men, trying to apply good, sound business principles to education. I think it would be a misfortune to leave it out. Our class room is not so much intended to give definite, concrete knowledge in mechanics. It has a far broader purpose, and the first purpose of all is to give these young men who are growing up and who are going to be our industrial army and our industrial non-commissioned officers, an objective as well as a subjective viewpoint. Our whole labor problem hinges on the fact that our men have only a subjective viewpoint and need an objective one, also; that they cannot see things from the other side as well, but only from their own side. In order to give them that objective viewpoint; in order to develop the character of the men; in order to make them well-intentioned men, good citizens, good working citizens, good workingmen, we have instituted, and I believe every manufacturer should institute, some class room work.

4 We must develop the intelligence at the same time that we are developing the hand. And it is not essential whether the instructor in our class room, pedagogically speaking, toes in or toes out when he is before his class.

TOPICAL DISCUSSION ON RECENT DEVELOPMENTS IN WHEEL TESTING

Dr. C. H. Benjamin

Before proceeding to describe the testing pit recently established at Purdue University for experimental work in bursting various rotating members by centrifugal force, it will perhaps be well to review the progress made in this kind of experimentation since the first work which was reported to the Society in 1898. At that time, the object was to determine the bursting speed of small model flywheels with different types of rims and joints.

2 The first wheels experimented on were 15 in. in diameter. They were rotated by means of a Dow steam turbine, the speed being measured by an electric commutator. The shield used was made of 2-in. pine plank, weighted with heavy castings and timbers. Fig. 1 shows the appearance of the shield after the first explosion. For succeeding experiments, a similar shield of 6 in. by 12 in. white oak was constructed.

3 An increase to 24 in. in the size of the wheels tested resulted in the complete wrecking of this shield, as may be seen by Fig. 2. In all further experiments conducted that year, the shield was made of oak timbers 12 in. square, firmly bolted together and covered with 3-in. oak plank. The speed of the flying fragments is indicated by the fact that some of them cut clean holes through moving belting, similar to those which would be made by bullets. During this year, ten 15-in. and seven 24-in. wheels were broken.

4 Further experiments on 24-in. model wheels were made during the following year and sixteen wheels were broken. The wheels in this series of experiments were enclosed in a cast-steel ring 36 in. in inside diameter, with a rim section 4 in. by 6 in. This was lined

This discussion was presented at St. Louis, March 12, 1910.

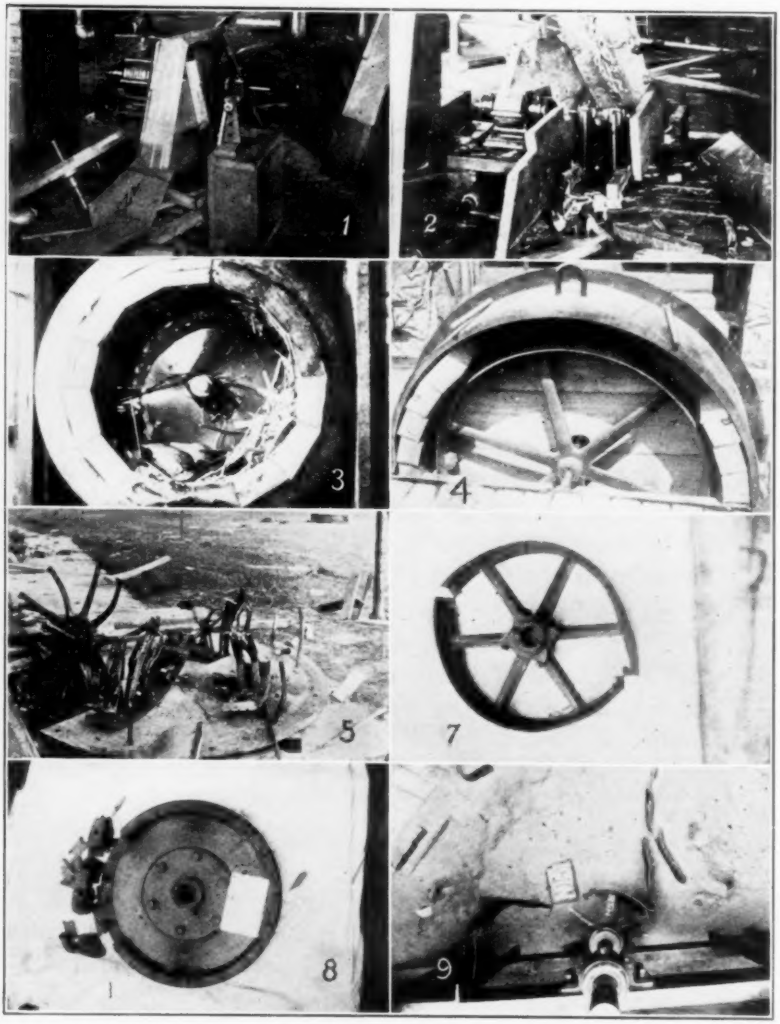


FIG. 1 SHIELD OF 2-IN. PINE AFTER EXPLOSION OF 15-IN. WHEEL
FIG. 2 SHIELD OF 6-IN. BY 12-IN. OAK AFTER EXPLOSION OF 24-IN. WHEEL
FIG. 3 CAST-STEEL RING AFTER EXPLOSION OF 24-IN. WHEEL
FIG. 4 STEEL SHIELD FOR FOUR 4-FT. PULLEYS
FIG. 5 WRECK OF 4-FT. WHEEL AND SHIELD
FIG. 7 24-IN. STEEL PULLEY SHOWING WEAK JOINT
FIG. 8 PAPER PULLEY
FIG. 9 VIEW LOOKING DOWN INTO PIT AFTER EXPLOSION OF A STEEL PULLEY

with wooden blocks to absorb the energy of the fragments, and was completely enclosed in oak planking. The same steam turbine was used for driving the pulleys, but a tachometer was used for indicating the speed.

5 The appearance of the casing and wheel after an explosion is shown in Fig. 3. After such an explosion, the wooden blocks would move around in the ring several inches, showing the tangential motion of the fragments. That this apparatus was not entirely safe was demonstrated in one experiment by the escape from the casing of portions of the pulley rim, due to the breaking of the retaining bolts. Although there were numerous spectators in the room, no accident occurred.

6 In this same apparatus fifteen emery wheels of different makes were burst, as reported to the Society in 1903. On account of the greater fragility of the emery wheels, no accident resulted.

7 These experiments were followed in the succeeding year by tests on wooden and steel pulleys 24 in. in diameter. The results of these experiments were published in August 1905 in Machinery. Eight pulleys were tested, including five wood split pulleys, one with steel arms, one all-steel pulley, and one wooden pulley with a solid web which we did not succeed in breaking. A number of pulleys made of paper fiber were tested in the same way, with results not very different from those on wooden pulleys.

8 In 1906 and 1907, a large number of cast-iron discs of various thicknesses and types of hub were exploded in the same apparatus. The results of these experiments, and the conclusions from them, have not yet been published.

9 It seemed desirable to test larger pulleys in the same manner, to see if the peripheral bursting speed would be the same for different sizes of pulleys. A few experiments were made in 1904, on pulleys 4 ft. in diameter. Former experiments had showed that it was hardly safe to burst pulleys of this weight and size inside a building. The apparatus shown in Fig. 4 was built entirely of steel and was 5 ft. in inside diameter. The shield was of rolled boiler plate, 1¼ in. thick and having a tensile strength of 65,000 lb. per sq. in. Flat plates ⅜ in. thick were bolted to the sides so as to enclose completely the wheel tested. Fig. 5 shows the shield after the explosion of the third wheel. The upper half of the casing, weighing about half a ton, was carried 75 ft. in the air and some hundred feet in a horizontal direction.

10 About this time the attention of the writer was called to a vertical shaft used by a German experimenter for bursting emery wheels,

the wheel being mounted at the lower end of the shaft inside a pit. The simplicity and safety of this form of construction are strong points in its favor. The last testing apparatus devised is constructed on this principle and located at Purdue University.

11 Fig. 6, showing a vertical section of the pit and connections, needs little explanation. The weight of the wheel and shaft is supported by a ball and thrust bearing, while rotation is effected by a 10 h.p. motor having a speed which can be varied from 800 to 2400 r.p.m. The pit itself is lined with concrete, but the impact of the fragments is received by a bank of sand. This works admirably to prevent any bruising or smashing of the fragments after the explosion. The speed is taken in the usual way by a tachometer.

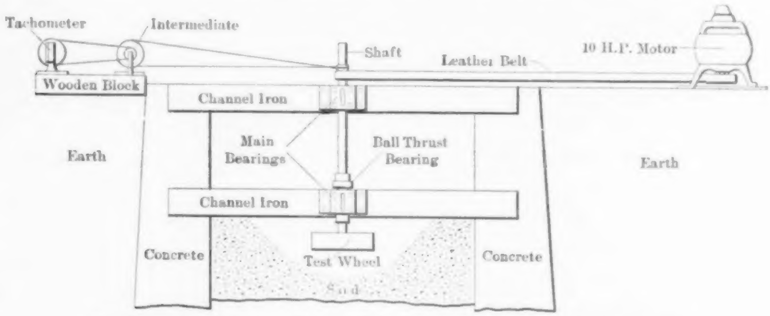


FIG. 6 FLYWHEEL TESTING PIT AT PURDUE UNIVERSITY

12 During the winter of 1908–1909, this apparatus was used very successfully[1] in testing the strength of sixteen pulleys, all 24 in. in diameter, with rims from 6 in. to 6¾ in. wide. The material used in their construction was wood, cast iron, paper and steel. Some of the rims were solid but most of them were of the usual split pulley type. The linear bursting speed of the solid wooden pulleys was about 275 ft. per sec., or 2600 r.p.m. The linear bursting speed of the split pulleys varied from 220 to 260 ft. per sec., or from 2100 to 2600 r.p.m. The paper pulleys on the other hand, having a solid web, were considerably stronger, averaging about 300 ft. per sec., linear bursting speed, or nearly 2900 r.p.m. Contrary to the usual opinion, the steel wheels are no stronger against bursting than the average wooden pulley. In fact, they are somewhat weaker than a well constructed pulley made of wood. Two wheels tested burst at exactly the same speed, 2240 r.p.m., or 235 ft. per sec.

[1] These experiments were made by Messrs. Biggs and Woodworth, senior students, as a part of their graduating theses.

13 The weakness of this type of pulley is due to the peculiar form of joint fastening, which is bent and broken by the centrifugal pressure. The bursting of the wooden pulleys was due in most cases to the greater density of the balance weights, consisting of slugs of round iron inserted in holes bored in the rim. These caused considerable centrifugal force at the points where they were located. It was evident from the appearance of the broken wheel that some of these weights had forced their way through the rim, thus starting rupture.

14 It is difficult to see how ordinary pulleys with wooden rims can be satisfactorily balanced without weakening. As there is rarely necessity for a linear speed of more than 100 ft. per sec., however, all of the pulleys tested had a factor of safety sufficient for commercial use. This is not true, however, of all pulleys. Two 4-ft. pulleys which were tested burst at speeds of 1100 and 600 r.p.m., respectively, which was considerably less than was expected of them. In the case of the pulley having a solid rim, this was due to the presence inside the rim of a balance weight of $3\frac{1}{2}$ lb. At 1100 r.p.m. the centrifugal force of this balance weight was over 2700 lb. In the same manner, 4-ft. pulley No. 2 was burst by the centrifugal pressure of a flange which weighed with its bolts $7\frac{1}{2}$ lb., and had a centrifugal force at bursting speed of nearly 1700 lb.

15 The effect of a joint flange is particularly disastrous, on account of the weakness of the joint itself to resist bending.

CONCLUSION

16 The bursting speed of most cast-iron pulleys having continuous rims may be put at about 400 ft. per sec., corresponding very nearly to a centrifugal tension of 16,000 lb. per sq. in. A wooden pulley with a continuous web and rim is even stronger than this, since wood is stronger in proportion to its weight than cast iron. A 2-ft. wooden pulley of this description has been run at a speed of 467 ft. per sec. without breaking. The ordinary split pulleys, whether of wood, steel or iron, cannot be relied upon at speeds much over 200 ft. per sec., on account of the weak points which have been mentioned. For experimental high speeds, steel pulleys of a much higher bursting point could undoubtedly be constructed. The poor joint design of the ordinary split steel pulley, such as is used for shafting transmission, renders it unusually weak in this respect.

17 It is proposed to use the testing pit for further experiments along several different lines, one being the testing of various kinds

of grinding wheels, including carborundum as well as the ordinary wet grindstone. The writer also hopes to test out several flywheel joints, on model wheels ranging from 4 ft. to 6 ft. in diameter; also to have some time an opportunity to test some band saw wheels in a similar manner.

DATA AND RESULTS OF EXPERIMENTS MADE IN THE NEW TESTING PIT

No. of Test	Kind of Material in Pulleys	Rim				Weight	Bursting Speed	
		Style	Diameter Inches	Breadth Inches	Depth Inches	Pounds	r. p. m.	Peripheral Speed Ft. per Sec.
1	wood	solid	24	6.25	1.62	29.37	2720	284.7
2	wood	solid	24	6.25	1.62	29.37	2550	266.9
3	wood	2 sections	24	6.5	1.78	29.67	2210	231.8
4	wood	2 sections	24	6.5	1.78	29.67	2110	220.8
5	wood	2 sections	24	6.5	1.78	28.81	2390	251.0
6	wood	2 sections	24	6.5	1.78	28.81	2430	254.3
7	wood	2 sections	24	6.5	1.78	28.81	2360	247
8	wood	2 sections	24	6.5	1.78	28.81	2420	253.3
9	wood	2 sections	24	6.5	1.78	28.81	2570	258.5
10	wood	2 sections	24	6.5	1.78	28.81	2535	244.4
11	cast iron	solid	24	6.0	0.406	70.44	3720	389.4
12	cast iron	solid	24	6.0	0.406	70.44	3380	353.8
13	paper	solid	24	6.0	1.75	77.37	2820	295.2
14	paper	solid	24	6.0	1.75	77.37	2930	306.7
15	steel	2 sections	24	6.75	0.0625	41.75	2240	234.5
16	steel	2 sections	24	6.75	0.0625	41.75	2240	234.5

FURTHER DISCUSSION

Following the introductory discussion by Dr. Benjamin, the question was asked by G. M. Peek if it had been found necessary, after the bursting of a pulley, to renew the driving shaft on which it had been mounted. Dr. Benjamin replied that the shaft was usually sprung by the unbalanced rotation of the pulley after bursting, and had to be replaced by a new one.

2 J. D. McPherson submitted the accompanying sketch (Fig. 1) of a large flywheel with heavy arms split on the centre of an arm. The two parts are dovetailed together and held by prisoners. He explained that this construction avoids the objectionable practice of placing the weight of the joint between the arms.

3 Dr. Benjamin stated that in this way any desired joint efficiency, up to 80 or 90 per cent, could be obtained. There can be no bending action due to centrifugal force and therefore no tendency to open the joint. Most split flywheels are now made with some form of joint over the arms.

4 H. A. Ferguson asked whether any experiments had been made on steel discs such as those used for the cold-sawing of structural steel. They are of open-hearth flange steel and run at a peripheral speed of 22,000 to 26,000 ft. per min. They are frequently replaced b cause of splits on the periphery, but this is apparently due to crystalization caused by the alternate heating and cooling to which they are subjected.

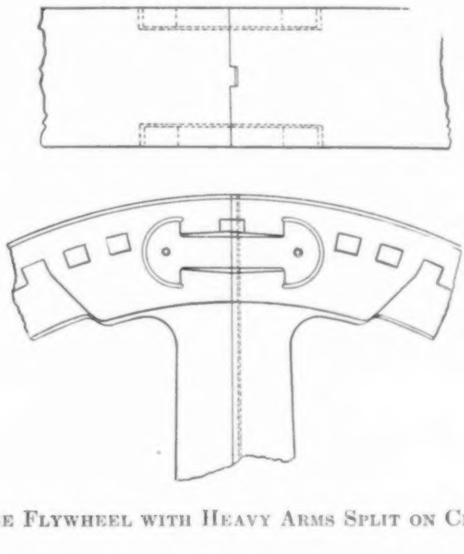


Fig. 1 Large Flywheel with Heavy Arms Split on Center of Arm

5 Dr. Benjamin said that little is known experimentally about the bursting speed of discs, but it is certain that they are stronger than rings, because the web resists the centrifugal tension. Stodola's Steam Turbines, the best theoretical work on the subject, states that the bursting strength of a ring is half that of a solid disc, but the speaker had never believed this statement, although it is theoretically correct. Solid steel discs are probably safe at the speed mentioned, but the practice of punching holes in the discs to balance them is questionable. Even a solid disc might be wrecked by centrifugal force in combination with the severe strains due to the cutting action.

6 Col. E. D. Meier suggested that, since flywheel problems deal with those of 10 or 12 ft. in diameter, and the expense of actual experiments on such wheels would be too great for private institutions, the engineering profession should use its influence to have such experiments made by the Government or at Government expense in private institutions.

7 Prof. H. Wade Hibbard announced that a series of experiments on aeroplane propellers were to be made this spring at the Missouri State University. The propellers are to be of wood, 6 ft. in diameter, and will be run at the speeds used in actual practice, i.e., 600 to 1800 r.p.m. It had been the intention to make the tests in the laboratory, but the experiences of Professor Benjamin with the bursting of flywheels would seem to indicate that this would be dangerous.

8 Dr. Benjamin responded that he would consider any such experiments extremely unsafe unless the apparatus were enclosed with some strong material. No matter what the material, when speeds of from 200 to 400 ft. per sec. are attained, it is vicious. In some of his experiments he found that fragments of wood went through a running belt, leaving holes as clean as a bullet would.

9 Colonel Meier suggested that Professor Hibbard's experiments be extended by testing some propellers to destruction in order to determine the factor of safety of those now being used.

GENERAL NOTES

AMERICAN SOCIETY OF CIVIL ENGINEERS

The annual convention of the American Society of Civil Engineers will be held in Chicago June 21-24, 1910.

At the regular monthly meeting on May 4, two papers were read: Water Supply of the El Paso Southwestern Railway from Carrizozo to Santa Rosa, New Mexico, by J. L. Campbell; and The New York Tunnel Extension of the Pennsylvania Railroad: The site of the Terminal Station, by G. C. Clarke. On May 18, J. C. Meem presented a paper entitled Pressure Resistance and Stability of Earth.

AMERICAN INSTITUTE OF MINING ENGINEERS

On Saturday evening, April 30, the American Institute of Mining Engineers gave a dinner in celebration of the seventieth birthday of Dr. Rossiter W. Raymond, for the last thirty years Secretary of the Institute. Nearly 400 were in attendance.

A gold medal from the Institution of Mining and Metallurgy was presented to Dr. Raymond by R. T. Bayliss, and illuminated parchments which were sent by various foreign engineering bodies were presented by E. G. Spillsburg, Mem. Am. Soc. M. E. Dr. Raymond was also made the recipient of a silver service at the close of the speaking.

Among those who paid tribute to Dr. Raymond were Dr. James Douglas; Dr. Lyman Abbott; George Westinghouse, President Am.Soc.M.Ee.; John Bensel; M. Sorzano de Tajada, of the Société des Ingénieurs Civils de France; Frank Dawson Adams, president of the Canadian Mining Institute; Robert W. Hunt, Past-President, Am.Soc.M.E.; Thomas Commerford Martin; William Lawrence Saunders, Mem.Am.Soc.M.E.

The guests were presented with an illustrated booklet containing scenes from the life of Dr. Raymond.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The annual convention of the American Insitute of Electrical Engineers will be held June 27-30, at the Waumbek Hotel and cottages at Jefferson, N. H., in the White Mountains. The annual meeting was held at New York May 17. Officers were declared elected as follows: president, Dugald C. Jackson, Mem. Am.Soc.M.E.; vice-presidents, Percy H. Thomas, H. W. Buck, Morgan Brooks, Mem.Am.Soc.M.E.; managers, H. H. Barnes, Jr., C. E. Scribner, W. S. Rugg, R. G. Black; treasurer, Geo. A. Hamilton; secretary, Ralph W. Pope.

On May 5 to 7 a meeting under the auspices of the High Tension Transmission Committee, of which Ralph D. Mershon, Mem.Am.Soc.M.E., is chairman, was held at San Francisco. Elaborate arrangements were made for the entertainment of the members, and professional papers were read as follows: Emergency Generating Stations for Service in Connection with Hydroelectric Transmission Plants under Pacific Coast Conditions, A. M. Hunt, Mem.Am. Soc.M.E.; Hydroelectric Power as Applied to Irrigation, J. C. Hays; The Developed High-Tension Net-Work of a General Power System, Paul M. Downing; Parallel Operation of Three-Phase Generators with their Neutrals interconnected, G. I. Rhodes; Observations of Harmonics in Current and Potential Wave Shapes of Transformers, John J. Frank; Transmission Line Crossings of Railroad Right-of-Way, A. H. Babcock.

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS

The stated annual meeting of the National Association of Cotton Manufacturers was held in the Mechanics Fair Building, Boston, Mass., April 27 and 28, 1910.

An address of welcome was made by Governor Draper of Massachusetts, to which Franklin W. Hobbs, of Boston, responded. Addresses followed by the president, Charles T. Plunkett, Mem.Am.Soc.M.E., Richard C. Maclaurin, President of the Massachusetts Institute of Technology, and Howard Ayres, secretary of the Cotton Goods Export Association. Papers were presented at the sessions of Wednesday afternoon and Thursday on The Progress of the Diesel Engine, Col. E. D. Meier, Mem.Am.Soc.M.E.; The Federal Corporation Tax Law, Walter S. Newhouse; A Substitute for Cotton, James Hope; Superheated Steam and Superheaters, Dr. D. S. Jacobus, Mem.Am.Soc.M.E.; The Electric Drive as a Manufacturing Proposition, Meldon H. Merrill; Choice of Power for Textile Mills, Charles T. Main, Mem.Am.Soc.M.E.; Recent Advances in the Chemistry of Coal Tar Colors, Dr. Hugo Schweitzer; Sizing of Vegetable Fibers, Hermann Seydel; Production-Increasing Methods, Henry L. Gantt, Mem. Am.Soc.M.E.; Distribution of Artificial Light, F. M. Scantlebury; Bibliography of the Cotton Manufacture, Dr. C. J. H. Woodbury, Mem.Am.Soc.M.E.

The following officers were elected: Franklin W. Hobbs, President; George Otis Draper and Edwin Farnham Greene, Vice-Presidents; Albert F. Bemis, R. M. Miller, Jr., Russell B. Lowe Frederick A. Flather, Mem. Am. Soc. M. E., and Frederick B. Macy, Directors.

LECTURES ON AËRIAL NAVIGATION AT MCGILL UNIVERSITY

The Department of Mechanical Engineering of McGill University has arranged a course of lectures on aërial navigation which will deal with the mechanical principles involved in the construction of the machines, the process of their manufacture, the difficulties of steering and manipulating with the various methods in use to obviate these troubles, and the displacement of air and the theory of gliding. The course will be in charge of Prof. C. M. McKergow.

ADVANCED STUDY OF ELECTRICAL ENGINEERING AT MASSACHUSETTS INSTITUTE OF TECHNOLOGY

The Massachusetts Institute of Technology will this year confer for the first time in its history the degree of Doctor of Engineering. Special attention will be given next year to graduate work. The lectures of Prof. Harold Pender will extend the discussion contained in his advanced lectures of this year on the high-voltage alternating transmission and utilization of power, with a repetition of the general treatment in his lectures of this year on the transmission circuit, and more attention will be given to the conditions arising from the utilization of power. Professor Jackson's lectures for graduate students on the organization and administration of public service companies will next year be directed more to the theory underlying methods of charging for service by public service companies, with particular reference to charges for electric light and power, but with collateral consideration of railroad and tramway charges and charges for gas and the service of other public utilities. Professor Wickenden will originate a course of lectures on illumination, photometry and illuminating engineering, as a part of the optional curriculum for undergraduate and graduate students.

FOREST PRODUCTS LABORATORY AT UNIVERSITY OF WISCONSIN

The Forest Service of the United States and the University of Wisconsin are coöperating in the establishment at Madison, Wis., of a Forest Products Laboratory, which will be prepared to carry on tests of the strength and other properties of timber, the preservative treatment of timber, the saving of wood waste by means of distillation processes, and the fiber of various woods for paper and other purposes. It is proposed to make it the largest and best equipped wood-testing laboratory in the world. The laboratory will be formally opened on June 4, 1910, and representatives of lumber manufacturing and wood-using associations from all parts of the country are expected to attend.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 12th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

026 Young technical graduate with good scholastic record and at least two years practical experience, for position of assistant in the laboratory of an engineering school; salary $1000 for the academic year. Location, Massachusetts.

027 Instructor in the mechanical engineering department of Columbia University, New York; will pay $1000 per year.

028 Manager of well-known company in New York State requires the services of an active, capable, educated and energetic man of good address, tactful in the management of men, familiar with approved systems of commercial and workshop management and costs. Candidate for such position should have thorough practical experience in machine shop and foundry producing heavy machinery for steel works, mills, etc.; large steam and gas engines, general heavy jobbing work and gas producers, also high-class marine and stationary boilers and heavy steel-plate work.

029 Engineer, experienced in the design of hydraulic machinery, pumps and large rolling mills for rolling sheet metal; must be thoroughly competent to make estimates and prepare complete calculations and data for drafting room. Location, Connecticut.

030 Michigan concern engaged in furnishing to several companies on cooperative basis, electric power, live steam, low pressure steam for heating, gas and compressed air, wishes to engage man of executive ability to take entire charge of plant and its operation; he must be qualified to give expert advice on changes or alterations, keep up output and give satisfactory service. Wants thoroughly high-grade competent man.

MEN AVAILABLE

73 University graduate, B.S.M.E., general experience with consulting engineer; five years in engineering and executive positions; building power plant machinery. Past six years with electric railway system in charge of design and construction of modern power house equipment, large units, high and low-pressure turbines, condensers, cooling towers, steam and gas engine plants. Will change on reasonable notice. Eastern location preferred.

74 Junior member, M.E., would like to connect with some consulting engineer, in or around New York. Industrial engineering preferred.

75 Young man desirous of getting away from close application to drafting board; technical graduate; experience in office and construction work as well as drafting, on power and industrial plants, wants commercial work with firm of engineers and contractors or with industrial company.

76 Engineer, 9 years experience in civil engineering, especially hydraulics; 5 years in charge of experimental steam turbine work; desires position as assistant professor of civil or mechanical engineering.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ABERCROMBIE, James Henderson (1901), Mech. Supt., Clark Thread Co., Clark and Ogden Sts., Newark, N. J.

ALBERGER, Louis R. (1889), Pres., Alberger Condenser Co., 140 Cedar St., New York, N. Y.

ATKINS, David Fowler (1907), Supv. Architects Office, Treasury Dept., Washington, D. C.

BIBBINS, James Rowland (1904; 1909), Engr. with Bion J. Arnold, 154 Nassau St., New York, N. Y.

BILLINGS, William Richardson (1906), Secy. and Treas., Alberger Condenser Co., and Alberger Pump Co., 140 Cedar St., New York, and 151 Columbia Hgts., Brooklyn, N. Y.

BIXBY, William P. (Junior, 1908), Mech. Dept., Erie R. R. Co., 50 Church St., New York, N. Y.

CAMPBELL, Gordon M. (1906), Genl. Elec. Co., Turbine Dept., West Lynn, Mass.

CASH, Arthur Wise (1899), Charge Regulating Valve and Engrg. Dept., H. Mueller Mfg. Co., Decatur, Ill.

CHAMBERS, Norman C. (Junior, 1905), Sales Dept., Niles-Bement-Pond Co., 111 Broadway, New York, N. Y., and *for mail*, care F. H. Bagge, Calle San Martin, 121, Buenos Aires, Argentine Republic, South America.

CREELMAN, Frank (1894), 447 W. 23d St., New York, and *for mail*, Hotel Cunningham, Sandy Hill, N. Y.

DAUGHERTY, Samuel Bovard (1905), Ch. Draftsman, Gas Eng. Dept., Snow Steam Pump Wks., and *for mail*, 129 N. Norwood Ave., Buffalo, N. Y.

DECKER, Edward P. (1906), E. P. Decker & Co., 80 Griswold St., and 79 Pingree Ave., Detroit, Mich.

ELLICOTT, Edw. Beach (1903), Elec. Engr., Sanitary Dist. of Chicago, 1500 Am. Trust Bldg., and *for mail*, 6229 Winthrop Ave., Chicago, Ill.

ESTES, William Wood (1891; 1904), Designer, Genl. Fire Extinguisher Co., and *for mail*, 245 Waterman St., East Side Sta., Providence, R. I.

FRANCIS, W. H. (1884), Union League, Broad and Sansom Sts., Philadelphia, Pa.

FRITZ, Aime L. G. (Junior, 1907), Ch. Draftsman, Hartford Suspension Co., 150 Bay St., Jersey City, N. J., and *for mail*, 99 Elmwood St., Woodhaven, L. I., N. Y.

GATES, Philetus W. (1902), Vice-President, 1906-1908; Pres., Hanna Engrg. Wks., 2059 Elston Ave., Chicago, Ill.

GIBBS, Geo. (1890), Ch. Engr. Elec. Traction and Terminal Sta. Constr., Pa. Tunnel & Terminal R. R. Co., Ch. Engr. Elec. Traction, West Jersey & Seashore R. R. Co., L. I. R. R. Co., 32d St. and Seventh Ave., New York, N. Y.

HAMERSTADT, William Diehl (Junior, 1907), Engr., Rockwood Mfg. Co., and *for mail*, 1608 Central Ave., Indianapolis, Ind.

HARTNESS, R. B. (Associate, 1903), 3042 Foster St., Los Angeles, Cal.

HECK, Robert C. H. (1906), Prof. Mech. Engrg., Rutgers College, and *for mail*, 35 College Ave., New Brunswick, N. J.

HEIKEL, Daniel August (1899), Life Member; M. E., Rm. 745, Oliver Bldg., Pittsburg, Pa.

HILL, E. Rowland (1907), Asst. to Ch. Engr., Elec. Traction, Pa. Tunnel & Terminal R. R., 32d St. and Seventh Ave., New York, N. Y., and 76 Watson Ave., East Orange, N. J.

HUNTER, John A. (1909), Steam Engr., Am. Sheet & Tin Plate Co., Frick Bldg., Pittsburg, and Dickson Ave., Ben Avon, Pa.

HUTTON, Mancius S. (Junior, 1908), Junior Salesman, Am. Radiator Co., Bundy Dept., 129 Federal St., and *for mail*, 172 Huntington Ave., Boston, Mass.

HVID, Rasmus M. (1907; Associate, 1909), 29 Market St., Bethlehem, Pa.

KEITH, Robert R. (Junior, 1904), Asst. Genl. Supt., Light Fuel Oil Pump Co., and *for mail*, 687 Farwell Ave., Milwaukee, Wis.

KNIGHT, Hervey S. (Associate, 1898), Pat. Lawyer and Expt., 726 Ninth St., N. W., and 30 Piney Branch Rd., Washington, D. C.

LANE, Henry Marquette (1900), Editor, Castings and Wood Craft, Caxton Bldg., and *for mail*, 10613 Greenlawn Ave., Cleveland, O.

LARKIN, A. C. (1895; 1905), Babcock & Wilcox, Ltd., College St., St. Henry, Montreal, Canada.

LEWIS, John Ernest (Junior, 1909), Paonia, Colo.

LILLIBRIDGE, Ray D. (Associate, 1907), 195 Broadway, and P. O. Box 824, New York, N. Y.

McARTHUR, Arthur Royal (1906), Resident Engr., Am. Sheet & Tin Plate Co., and *for mail*, 674 Harrison St., Gary, Ind.

MARBURG, Louis Chas. (1909), 1777 Broadway, New York, N. Y.

MORLEY, Ralph (Junior, 1906), Mech. Engr., Transmission Dept., The Fairbanks Co., New York, N. Y., and *for mail*, 153 Delavan Ave., Newark, N. J.

NEWCOMB, Chas. L., Jr. (Associate, 1908), Denver Rock Drill & Mchy. Co., 18th and Blake Sts., Denver, Colo.

PARSONS, W. Everett (1899), Cons. Engr., 12 Bridge St., New York, and *for mail*, 10 Rich Ave., Mt. Vernon, N. Y.

PERRY, Samuel B. (Junior, 1895), Ins. Engr., 68 William St., New York, and Hollis, L. I., N.Y.

PHELPS, Charles C. (Junior, 1909), Editor, Steam, 2108 West St. Bldg., New York, N. Y.

POSEY, James (Junior, 1907), Cons. Engr., Painter & Posey, Cons. Engrs., 324 N. Charles St., Baltimore, Md.

RAY, Frederick (Junior, 1903), Ch. Engr., Alberger Pump Co., 140 Cedar St., New York, N. Y., and 19 Wilcox Place, East Orange, N. J.

REEDER, Nathaniel S., Jr. (1902; 1907), West Steel Car & Fdy. Co., 1470 Old Colony Bldg., Chicago, Ill.

ROGERS, Robert W. (Junior, 1908), Erie R. R., and *for mail*, 216 Walnut St., Meadville, Pa.

SALTZMAN, Auguste L. (1908), M.E., Asst. Ch. Engr., Edison Cos., Edison Laboratory, Orange, and *for mail*, 41 Watson Ave., East Orange, N. J.

SAMPSON, Chas.C. (1909), M. E., Supt. Constr., 5-8 Blowing Eng., Illinois Steel Co., South Chicago, and *for mail*, 7318 Champlain Ave., Chicago, Ill.
SANDO, Will J. (1899), Manager, 1908-1911; 430 Kane Pl., Milwaukee, Wis.
SCOTT, Walter G. (Junior, 1909), Cyclone Drill Co., Orville, O.
SEYMOUR, Dudley S. (1905), Supt., Union Spec. Mch. Co., 300 W. Kinzie St., Chicago, and *for mail*, 228 N. Elmwood Ave., Oak Park, Ill.
SMITH, Harry Ernest (1897), Chem. and Engr. of Tests, L. S. & M. S. Ry., Collinwood, and *for mail*, 36 Beersford Place, East Cleveland, O.
SMITH, J. Waldo (1896), Ch. Engr., Board of Water Supply, City of N. Y., 165 Broadway, and 136 Madison Ave., New York, N. Y.
STEENSTRUP, Peter Severin (1906), Box 1843, Seattle, Wash.
SYMONDS, George P. (1908), Chief Engrg. Dept., Alberger Condenser Co., 140 Cedar St., New York, N. Y.
TADDIKEN, J. F., Jr. (Junior, 1907), Am. Beet Sugar Co., Chino, Cal.
TERWILLIGER, Harry L. (Associate, 1901), Sales Mgr., Harron, Rickard & McCone, 139-149 Townsend St., San Francisco, and *for mail*, 1121 Emerson St., Palo Alto, Cal.
ULRICH, Max Julius (1906), Ch. Draftsman, Alberger Condenser Co., 140 Cedar St., New York, N. Y.
WADSWORTH, Frank L. O. (1903), Cons. Engr., 1347-1348 Oliver Bldg., and Duquesne Club, Pittsburg, Pa.
WESTERFIELD, George Sumner (Junior, 1903), Mgr. and Dist. Mgr., Hooven, Owens, Rentschler Co., Warren Webster & Co., B. F. Sturtevant Co., 326-329 Hennen Bldg., and 1320 Eleonore St., New Orleans, La.
WHITE, James A. (Junior, 1900), Genl. Elec. Co., and *for mail*, 19 Red Rock St., Lynn, Mass.
WHITEFORD, James F. (1908), Santa Fe Shops, Topeka, Kan.
WILLIAMSON, Leroy A. (Associate, 1902), Board of Trade Bldg., 131 State St., Boston, Mass.
WILSON, Wm. R. (Junior, 1899), Alberger Condenser Co., 140 Cedar St., New York, and *for mail*, 224 Palisade Ave., Yonkers, N. Y.
WINSHIP, James G. (1891), Internatl. Steam Pump Co., 115 Broadway, New York, and *for mail*, 209 Ocean Ave., Brooklyn, N. Y.

NEW MEMBERS

HODGE, Wm. W. (Junior, 1909), Field Engr., Dodge & Day, Lewiston, Pa.

DEATHS

BARY, Mark, December 1909.
BLOOMBERG, Jonas H.
EMERSON, Ralph Waldo, April 13, 1910.
PARSONS, William N., April 24, 1910.
PLUMMER, Frank J., April 15, 1910.
SPARROW, Ernest P., April 18, 1910.

GAS POWER SECTION

CHANGES OF ADDRESS

BIBBINS, James Rowland (1908), Mem.Am.Soc.M.E.
BIGELOW, Lucius S. (Affiliate, 1910), Pres., Light Pub. Co.,Pres.,Periodical's Pub. Co., 125 S. Main St., Willimantic, Conn.
HILLEBRAND, Herman (Affiliate, 1909), 638 W. Broad St., Bethlehem, Pa.
HOPCROFT, Ernest Bigly (Affiliate, 1908), L. W. Hall & Co., 50 Congress St., Boston, Mass.
MORLEY, Ralph (1908), Mem. Am.Soc.M.E.
RALSTON, Louis C. (Affiliate, 1909), R. F. D. 21, Box 41 A, San Jose, Cal.
SAMPSON, Chas. C. (1909), Mem.Am.Soc.M.E.

NEW MEMBERS

RIEPPEL, Paul (Affiliate, 1910), Blohm & Voss, Hamburg, Germany.

DEATHS

SPARROW, Ernest P., April 18, 1910.

STUDENT BRANCHES

CHANGES OF ADDRESS

GOLDSMITH, W. M. (Student, 1909), Greenwood Court, Greenwood Ave., Avondale, Cincinnati, O.
HESS, Harry L. (Student, 1909), Marysville, Cal.
LEVY, M. S. (Student, 1909), Metropole Hotel, Chicago, Ill.
MUDD, John P. (Student, 1909), 229 Zeralda St., Philadelphia, Pa.
SHULTS, L. J. (Student, 1909), 1820 S. Sawyer Ave., Chicago, Ill.
THOMAS, W. E. (Student, 1909), 4028 Sheridan Rd., Chicago, Ill.
WATSON, R. D. (Student, 1910), 237 Langdon St., Madison, Wis.

NEW MEMBERS

STEVENS INSTITUTE OF TECHNOLOGY

POLHEMUS, D. A. (Student, 1910), Stevens Inst. of Tech., Hoboken, N. J.

UNIVERSITY OF MAINE

BLAISDELL, A. H. (Student, 1910), 57 Fifth Ave., Bangor, Me.
COLE, R. F. (Student, 1910), Phi Beta Kappa House, Orono, Me.
CUMMINGS, C. G. (Student, 1910), Delta Tau Delta House, Orono, Me.
DANFORTH, H. N. (Student, 1910), Alpha Tau Omega House, Orono, Me.
HAMMOND, A. C. (Student, 1910), Main St., Orono, Me.
HARDY, S. J. (Student, 1910), Delta Tau Delta House, Orono, Me.
JOHNSON, C. A. (Student, 1910), Orono, Me.
LITTLEFIELD, P. H. (Student, 1910), Orono, Me.
MERRIAM, F. E. (Student, 1910), Orono, Me.
SCALES, E. M. (Student, 1910), Theta Epsilon House, Orono, Me.
SIMONTON, P. D. (Student, 1910), Orono, Me.

UNIVERSITY OF MISSOURI

EDGAR, O. N. (Student, 1910), 605 S. Fourth St., Columbia, Mo.
KENNEDY, F. T. (Student, 1910), Benton Hall, Columbia, Mo.
OLSEN, C. A. (Student, 1910), 411 S. Fifth St., Columbia, Mo.
PHILLIPS, E. C. (Student, 1910), 505 Conley Ave., Columbia, Mo.
PRICE, H. W. (Student, 1910), Lowry Hall, Columbia, Mo.
SEXTON, C. E. (Student, 1910), 605 S. Fourth St., Columbia, Mo.
SHARP, H. N. (Student, 1910), 311 Waugh St., Columbia, Mo.
STEED, A. (Student, 1910), Benton Hall, Columbia, Mo.
THACHER, F. B. (Student, 1910), Y. M. C. A. Bldg., Columbia, Mo.
WEAVER, H. E. (Student, 1910), 803 Virginia Ave., Columbia, Mo.
WESTCOTT, A. L. (Student, 1910), 1402 Windsor St., Columbia, Mo.

UNIVERSITY OF WISCONSIN

FALK, G. S. (Student, 1910), 627 Lake St., Madison, Wis.

COMING MEETINGS

May–June

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN EXPOSITION IN BERLIN

June 1–Aug. 31. American Manager, Max Vieweger, 50 Church St., New York.

AMERICAN BRASS FOUNDERS' ASSOCIATION

June 6–10, Detroit, Mich. Papers: Costs and Cost Systems, C. R. Stevenson; Analysis for Lead in Brass Alloys, C. P. Karr; Coöperative Course in Metallurgy, J. J. Porter; Electric Furnaces for Melting Non-Ferrous Alloys, A. L. Marsh; Fluxes as Applied to the Brass Foundry, I. S. Sperry; Electric Power as Applied to Melting, J. W. Richards, Mem. Am. Soc. M. E. Secy., W. M. Corse, Lumen Bearing Co., Buffalo, N. Y.

AMERICAN FOUNDRYMEN'S ASSOCIATION

June 6–10, Detroit, Mich. Secy. of general committee, A. Preston Henry, Standard Pattern Works.

ASSOCIATION OF CAR-LIGHTING ENGINEERS

June 7–8, semi-annual convention, Buffalo, N. Y. Secy., Geo. B. Colegrave, care of Central Railway, Chicago.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

June 22–24, summer meeting, Niagara Falls, N. Y. Secy., J. C. Olsen, Polytechnic Inst., Brooklyn.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

June 27–28, Annual Convention, Waumbek Hotel, Jefferson, N. H. Secy., R. W. Pope, 33 W. 39th St.

AMERICAN PORTLAND CEMENT MANUFACTURERS

June, Kansas City, Kans. Secy., P. H. Wilson, Land Title Bldg., Philadelphia, Pa.

AMERICAN RAILWAY ACCOUNTING OFFICERS

June 29, Colorado Springs, Colo. Secy., C. G. Phillips, 143 Dearborn St., Chicago.

AMERICAN RAILWAY MASTER MECHANICS ASSOCIATION

June 20–22, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

AMERICAN SOCIETY OF CIVIL ENGINEERS

June 1, 220 W. 57th St., New York. Papers: The New York Tunnel Extension of the Pennsylvania Railroad, B. H. M. Hewett; The North River Tunnels, W. L. Brown. June 21–24, Annual Convention, Chicago, Ill. Secy., C. W. Hunt, New York.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
May 27, St. Louis, with coöperation of Engineers Club of St. Louis. May 31–June 3, Spring Meeting, Atlantic City, N. J. July 26–29, meeting with Institution of Mechanical Engineers, in Birmingham and London, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SOCIETY FOR TESTING MATERIALS
June 28–July 2, annual meeting, Atlantic City, N. J. Secy., Edgar Marburg, University of Pennsylvania, Philadelphia.

CANADIAN ELECTRICAL ASSOCIATION
July 6–8, annual convention, Royal Muskoka, Lake Rosseau. Secy., T. S. Young, Confederation Life Bldg., Toronto, Ont.

CANADIAN GAS ASSOCIATION
June 9–11, annual convention, Alexandra Rink, Hamilton, Ont. Secy., A. W. Moore, Woodstock, Ont.

CLEVELAND ENGINEERING SOCIETY
June 14, annual meeting, 714 Caxton Bldg. Secy., J. C. Beardsley.

ENGINEERS' CLUB OF BALTIMORE
June 4, annual meeting. Secy., R. K. Compton, City Hall.

ENGINEERS SOCIETY OF MILWAUKEE
June 8, annual meeting, Builders Club. Secy., W. F. Martin, 456 Broadway.

ENGINEERS SOCIETY OF PENNSYLVANIA
June 7, annual meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher, P. O. Box 704.

FREIGHT CLAIM ASSOCIATION
June 15, Los Angeles, Cal. Secy., W. P. Taylor, Richmond, Va.

INTERNATIONAL CONGRESS OF INVENTORS
June 13–18, Rochester, N. Y.

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY
Last week in June, Düsseldorf, Prussia. Secy., Dr. E. Schrödter, Jacobstrasse 315.

IOWA DISTRICT GAS ASSOCIATION
June 15–17, annual meeting, Sioux City. Secy., G. I. Vincent, Des Moines.

MANUFACTURERS' SUPPLY ASSOCIATION
June 6–10, exhibit, Detroit, Mich. Secy., C. E. Hoyt, Lewis Institute, Chicago.

MASTER CAR BUILDERS ASSOCIATION
June 15–17, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

NATIONAL DISTRICT HEATING ASSOCIATION
June 1–3, annual meeting, Toledo, O. Secy., D. C. Gaskill, Greenville, O.

NATIONAL ELECTRIC CONTRACTORS' ASSOCIATION
July 20, annual meeting, Atlantic City, N. J. Secy., W. H. Morton, Martin Bldg., Utica, N. Y.

NATIONAL ELECTRIC TRADES ASSOCIATION
June, San Francisco, Cal. Secy., F. B. Vose, 1343 Marquette Bldg., Chicago.

NATIONAL GAS AND GASOLINE ENGINE TRADES ASSOCIATION

June 13–16, semi-annual meeting, Hotel Sinton, Cincinnati, O. Subjects for discussion: Carbureters, Geo. M. Schebler; Ignition in Gas and Gasoline Engines, Carl Pfanstiehl; Gas Producers, L. F. Burger; Present and Future Opportunities in the South for the Gas Engine Boiler, W. R. C. Smith; How to Illustrate and Describe Mechanical Installations, O. Monnett; Heavy Hitting in the Advertisers' League, Ren Mulford, Jr.; Some Association Experiences, F. J. Alvin; Dry Batteries, H. S. Green; The Gas Engine Field in Mexico, G. W. Hall; Large Gas Engines, J. D. Lyon. Secy., Albert Stritmatter.

NATIONAL SOCIETY FOR PROMOTION ENGINEERING EDUCATION

June 23–25, annual meeting, Madison, Wis. Papers on Technical Education Abroad; Inspection Trips for Technical Students, Efficiency in Technical Education. Secy., Prof. H. H. Norris, Cornell University, Ithaca, N. Y.

NEW ENGLAND WATERWORKS ASSOCIATION

June, Providence, R. I. September 14–16, annual convention, Rochester N. Y. Secy., Willard Kent, Narragansett Pier, R. I.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

June 28, annual meeting. Secy., T. M. Phetteplace, Mem.Am.Soc.M.E., 48 Snow St.

RAILWAY SIGNAL ASSOCIATION

June 14, 29 W. 39th St., New York, 9.30 a.m. Secy., C. C. Rosenberg, Bethlehem, Pa.

RENSSELAER SOCIETY OF ENGINEERS

June, annual meeting, Rensselaer Polytechnic Inst., 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

TELEPHONE SOCIETY OF NEW YORK

June 21, annual meeting, 29 W. 39th St. Secy., T. H. Woolhouse.

TRANSPORTATION AND CAR ACCOUNTING OFFICERS

June 28. Secy., G. P. Conard, 24 Park Pl., New York.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
June			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
3	Western Union Electrical Society	H. C. Northen	7.00
4	Amer. Soc. Hun.Engrs and Archts	E. L. Mandel	8.30
9	Illuminating Engineering Society	P. S. Millar	8.15
10	Western Union Electrical Society	H. C. Northen	7.00
			a.m.
14	Railway Signal Association	C. C. Rosenberg	9.30
			p.m.
17	Western Union Electrical Society	H. C. Northen	7.00
21	New York Telephone Society	T. H. Lawrence	8.00
24	Western Union Electrical Society	H. C. Northen	7.00
July			
7	Blue Room Engineering Society	W. D. Sprague	8.00

OFFICERS AND COUNCIL

PRESIDENT

George Westinghouse .... Pittsburg, Pa.

VICE-PRESIDENTS

Geo. M. Bond .... Hartford, Conn.
R. C. Carpenter .... Ithaca, N. Y.
F. M. Whyte .... New York

Terms expire at Annual Meeting of 1910

Charles Whiting Baker .... New York
W. F. M. Goss .... Urbana, Ill.
E. D. Meier .... New York

Terms expire at Annual Meeting of 1911

PAST PRESIDENTS

Members of the Council for 1910

John R. Freeman .... Providence, R. I.
Frederick W. Taylor .... Philadelphia, Pa.
F. R. Hutton .... New York
M. L. Holman .... St. Louis, Mo.
Jesse M. Smith .... New York

MANAGERS

Wm. L. Abbott .... Chicago, Ill.
Alex. C. Humphreys .... New York
Henry G. Stott .... New York

Terms expire at Annual Meeting of 1910

H. L. Gantt .... Pawtucket, R. I.
I. E. Moultrop .... Boston, Mass.
W. J. Sando .... Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

J. Sellers Bancroft .... Philadelphia, Pa.
James Hartness .... Springfield, Vt.
H. G. Reist .... Schenectady, N. Y.

Terms expire at Annual Meeting of 1912

TREASURER

William H. Wiley .... New York

CHAIRMAN OF THE FINANCE COMMITTEE

Arthur M. Waitt .... New York

HONORARY SECRETARY

F. R. Hutton .... New York

SECRETARY

Calvin W. Rice .... 29 West 39th Street, New York

EXECUTIVE COMMITTEE OF THE COUNCIL

Alex. C. Humphreys, *Chairman*
Chas. Whiting Baker, *Vice-Chairman*
F. R. Hutton
H. L. Gantt
F. M. Whyte

STANDING COMMITTEES

FINANCE

Arthur M. Waitt (5), *Chairman*
Robert M. Dixon (3), *Vice-Chairman*
Edward F. Schnuck (1)
Geo. J. Roberts (2)
Waldo H. Marshall (4)

HOUSE

William Carter Dickerman (1) *Chairman*
Francis Blossom (3)
Bernard V. Swenson (2)
Edward Van Winkle (4)
H. R. Cobleigh (5)

LIBRARY

John W. Lieb, Jr. (3), *Chairman*
Leonard Waldo (2)
Ambrose Swasey (1)
Chas. L. Clarke (4)
Alfred Noble (5)

MEETINGS

Willis E. Hall (5), *Chairman*
L. R. Pomeroy (2)
Wm. H. Bryan (1)
Chas. E. Lucke (3)
H. de B. Parsons (4)

MEMBERSHIP

Charles R. Richards (1) *Chairman*
George J. Foran (3)
Francis H. Stillman (2)
Hosea Webster (4)
Theo. Stebbins (5)

PUBLICATION

D. S. Jacobus (1) *Chairman*
Fred R. Low (3)
H. F. J. Porter (2)
Geo. I. Rockwood (4)
Geo. M. Basford (5)

RESEARCH

W. F. M. Goss (4), *Chairman*
R. H. Rice (2)
R. C. Carpenter (1)
Ralph D. Mershon (3)
Jas. Christie (5)

Note—Numbers in parentheses indicate number of years the member is yet to serve.

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus
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On Constitution and By-Laws

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On International Standard for Pipe Threads

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Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	John S. Langwill	Karl L. Kraatz
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Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	M. E. Strieter	A. D. Stancliff
	1910			
Univ. of Maine, Orono, Me.	February 8	Arthur C. Jewett....	H. N. Danforth	A. H. Blaisdell
Univ. of Arkansas, Fayetteville, Ark.	April 12	B. N. Wilson	C. B. Boles	W. Q. Williams